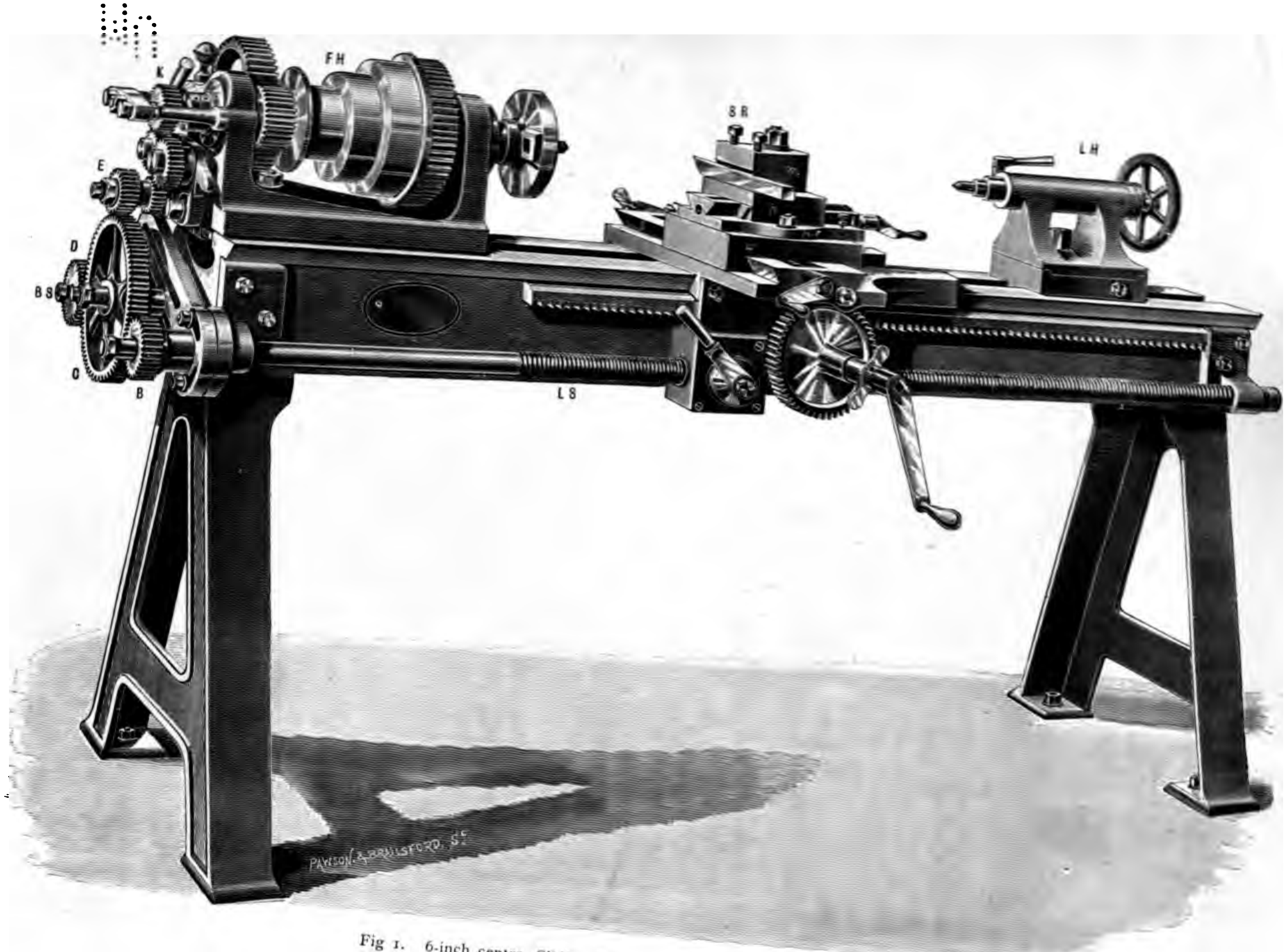


Fig 1. 6-inch centre, Sliding, Surfacing, and Screw-cutting Lathe.

A COURSE OF INSTRUCTION IN

MACHINE DRAWING & DESIGN

FOR TECHNICAL SCHOOLS AND ENGINEER STUDENTS

BY

WILLIAM RIPPER, M. INST. C.E.

PROFESSOR OF ENGINEERING IN THE FIRTH COLLEGE, SHEFFIELD
AND PRINCIPAL OF THE SHEFFIELD TECHNICAL SCHOOL

WITH FIFTY-FIVE PLATES AND NUMEROUS EXPLANATORY ENGRAVINGS

SECOND THOUSAND

RIVINGTON, PERCIVAL & CO.
KING STREET, COVENT GARDEN
LONDON
1896

PREFACE.

THE present work is intended to provide a course of practice and instruction in Machine Drawing and the elements of Machine Design for technical schools and engineer students, which shall be preparatory or supplementary to that special training to be obtained only in the works and office of the commercial engineer.

The special features of the work are :—

(1) A series of carefully graduated exercises, extending over fifty-two plates, commencing with simple projection, and advancing to the more difficult examples occurring in actual practice, and including engine and machine details, wheel gearing, slide valve gearing, expansion valves, the link motion, the screw propeller, &c. The student is required in every case to work to the dimensions, which are given in full throughout, and on no account merely to copy. Usually the scale of the sketch is such as to make mere copying or proportionate enlarging somewhat difficult. The method of providing incomplete and intentionally incorrect sketches has been avoided, such exercises being, in the author's opinion, more suitable as tests of knowledge already acquired, than as a means of acquiring such knowledge, and tending rather to increase than to remove a young student's difficulties.

(2) A large number of the plates are accompanied by explanatory sketches, which it is hoped will greatly assist the student to comprehend the true character and form of the thing represented, and thus remove a difficulty by which beginners are met at the threshold of the subject. The best explanation of the drawing of a machine is, no doubt, the actual machine itself, or a model of it; but these aids are not always possible, and where they cannot be obtained it is believed the help afforded by the sketches will be appreciated.

(3) For the earlier exercises, a small, simple engine, fitted with link motion reversing gear, has been chosen, and is such as shall be well within the scope of an ordinary student's powers. He is expected to make a complete set of working drawings of the details of the engine, and afterwards, from his own drawings, to build up and complete the General Drawing. A similar course is provided for the lathe, which is fitted with self-acting arrangements for sliding, surfacing, and screw-cutting. The engine is further intended to be used as a basis for exercises in design, by converting it, by variations or additions, into any required type.

(4) The plates are accompanied with useful information on the construction and design of the thing drawn, which may be studied with advantage concurrently with the exercise in drawing.

(5) Preliminary chapters are added on the making of working drawings, on shading and colouring finished drawings, on the strength of materials, and the principles of construction, with numerous illustrations and exercises.

In the preparation of the work I have gratefully to acknowledge the assistance received from the many sources of information at the disposal of the engineer, and especially from the works of Professors RANKINE, UNWIN, and REULEAUX, and Mr. SEATON.

I wish also to express my indebtedness to my brother, Mr. CHARLES RIPPER, for his very valuable help in the preparation of the plates; and to Messrs. PAWSON and BRAILSFORD (Engravers and Lithographers) and their able staff, for the ready and efficient way in which I have been assisted by them.

WILLIAM RIPPER.

THE TECHNICAL SCHOOL,
SHEFFIELD,
1888.

CONTENTS.

CHAPTER I.

ERRATA.

Page 16, lines 6 and 7, for 20126 lbs. = 9 tons,
read 12271 lbs. = $5\frac{1}{2}$ tons, nearly.

Page 53, line 14, for ·3, read ·3 *from centre of bolt.*

Plate IV., Fig. 2, vertical dimension $1\frac{3}{8}$ in. The lower arrow-head should be moved to line above.

CHAPTER IV.

CHAPTER V.

CHAPTER VI.

ENGINE CONSTRUCTION.

CHAPTER VII.

BOILER WORK.

CHAPTER VIII.

CHAPTER IX.

MACHINE DRAWING AND DESIGN.

Chapter I.

INTRODUCTION.

MACHINE DRAWING has been well described as "the language of the workshop," for by its aid it is possible to convey to the mind of a skilled workman clear and exact information as to the form and dimensions of any object we may require to have constructed, however complex and intricate that object may be. The power to make and to understand such a drawing is evidently, therefore, of great importance to all persons engaged in manufacturing industry, whether as constructors, users, buyers, or sellers of machinery.

Any drawing which conveys fully and clearly the idea of its author, accomplishes the end for which it is intended, and is, therefore, for all practical purposes, a good drawing, even though it be merely a "rough and ready sketch" in pencil. But it is hardly the novice who can make such a sketch; it is rather the man who can, when necessary, make an accurate and carefully-finished drawing—such as might be placed before a Board of Directors with an estimate—who knows exactly what to show on a sketch and how to show it.

Practice in machine drawing does more for the student than merely teach him to use his instruments skilfully, or to properly represent objects by plan, elevation, and section; for he is at the same time acquiring an intimate knowledge of the construction of the thing drawn. Henceforth the observant student who has been drawing, say an engine connecting rod, compares every new connecting rod he sees with the one he has drawn. The eye sees what it possesses the power to see, and every new example with which he becomes thoroughly familiar, by drawing it, enlarges his power of vision, and enables him to see with advantage many things which would otherwise pass before him unobserved.

Mechanical drawings are of three kinds:—First, *Detail Drawings*, which show to as large a scale as possible the form and dimensions of the various separate details of the machine. Second, *General Drawings*, which show plans, elevations, and sections of the machine as a whole, drawn to

a small scale. In these drawings all the parts are shown in position, and important details hidden from view are shown by dotted lines. Third, *Finished Drawings*, on which dotted work is omitted, and the drawings are more or less highly finished with shade lines or colouring.

In the drawing-office of a mechanical engineer, when a design is required for a special engine or machine, a rough approximate drawing or sketch is prepared, showing the general style and arrangement proposed. When this has been approved, a general drawing is then made to scale, each detail being carefully designed and placed in position, care being taken that the moving parts work quite freely and without danger of coming in contact with other parts. From this drawing, separate large drawings are made of the various details. These are traced (or copied by some process) on tracing cloth or paper, and passed forward into the workshop to act as guides to the workmen in the construction of the machine.

In the present work this order of procedure is reversed, the student beginning by making working drawings of simple details, advancing by stages to more difficult examples, and finally building up from the details already made, the general drawing of the engine or machine. The student is then in a position to apply himself to an original design, assisted by the notes given, and by such other help as he can obtain from actual examples and other sources.

2. DRAWING APPARATUS.

The materials required are: a drawing board, T square, two set squares, a box of instruments, drawing paper, pins, india-rubber, pencils, a rule, a set of scales, Indian ink, colours, saucers for mixing colours.

THE DRAWING BOARD.—The size of the drawing board will depend upon the size of the drawings to be made. In the engineer's office it is usual to make machine drawings as large as possible, as such drawings are far more acceptable to the workman and more easily understood by him than small closely-lined drawings. The board is made a little larger than the paper it is intended to take. For young students a board 16 ins. × 12 ins., to take half an imperial sheet of paper, is a suitable size. Drawings done on imperial sheets and on boards 31 ins. × 23 ins. are preferable.

The following are the names and sizes of drawing paper :—

	INCHES.
Demy	20 × 15½
Medium	22 × 17½
Royal	24 × 19½
Imperial	30 × 22
Atlas	34 × 26
Double Elephant...	40 × 27
Antiquarian	52 × 31

The size commonly used in an engineer's office is the double elephant. There are two kinds of surfaces—smooth and rough. The smooth is best for fine lining, but the rough surface takes colour best. Cartridge paper, a cheap, useful paper, can be obtained in sheets of various sizes, or in rolls.

THE T SQUARE should be long enough to reach across the board lengthways.

THE SET SQUARES, which are triangular pieces made of pearwood or ebonite, should be,—one having angles of 45° and another angles of 60° and 30°. Their smaller edges should be about 6ins. long.

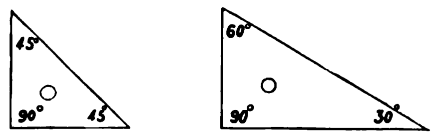


Fig. 2.

THE BOX OF INSTRUMENTS should be of as good quality as the student can afford to buy. It is a great mistake to suppose that inferior instruments are "good enough" for a beginner. The set should consist of:—A pair of large compasses with pen and pencil legs and lengthening bar, small pen and pencil bows, one or two drawing pens. Other useful additions are:—A pair of hair dividers with screw adjustment, a set of three small spring bows, including points, pen, and pencil. Instruments with needle points and double joints are the best.

THE PENCILS most suitable for ordinary work are those marked HH. For very fine work those marked HHH are better. When sharpened with a *flat* or *chisel point* the pencil retains its edge longer than when sharpened to a point. For many purposes, however, the point is preferable. To get a good edge or point, use a little piece of fine sand-paper.

DRAWING PINS are used to secure the paper for ordinary work which is to be got out of hand quickly; but for drawings upon which much labour is to be spent the paper should be stretched upon the board.

A BOXWOOD "TWO-FOOT" RULE, and a boxwood or ivory SCALE marked with scales from $\frac{1}{8}$ in. to 3 ins. to the foot, are necessary. The rule when made double folding is convenient for the pocket. Sets of cardboard scales may be obtained, which are cheaper than the boxwood scale.

COLOURS.—Indian ink, Prussian blue, neutral tint or Payne's grey, crimson lake, gamboge, burnt umber, yellow ochre, burnt sienna, light red, carmine.

Saucers for mixing ink or colours are sold in sets or "nests." They fit over each other, and when so placed the ink or colour is kept moist for a very long time.

Sable hair brushes are the best.

3. GENERAL INSTRUCTIONS.

TO STRETCH A SHEET OF PAPER.—First melt some glue in an ordinary glue pot, and let it be made thin enough to run freely. Lay the paper flat, and wet the surface opposite to that upon which the drawing is to be made all over with a sponge or clean towel. See that the moisture is evenly distributed over the paper, and keep it wet for about ten minutes. Turn the paper right side upwards. [Or, where there are suitable conveniences, dip the whole sheet of paper in a bath of clean water, and allow it to remain about five minutes; take it out, and lay it carefully on the board with the right side up.] Turn up the edges all round about $\frac{3}{4}$ in., and dry the edge by wiping it with a towel. Now apply the glue, not too thickly, to a long edge of the paper, and rub it down on the board with the smooth end of a knife handle. Treat the opposite long edge in the same way, and then the two short edges. Allow the board to lie flat till the paper is quite dry, when it will be found to be tightly stretched upon the board.

Directions for the Preparation of Drawings.

1. The T square should be held by the head, and not by the blade, and it should be moved by the left hand on the left hand edge of the board only. All horizontal lines are drawn with the T square, and all vertical lines with a set square, which works on the edge of the T square.

2. Drawing in pencil. Lines are drawn from right to left, or from the bottom towards the top of the board. Lines which represent the outline or boundary of an object are shown as *full* lines. Lines which represent parts which are hidden from view are shown as *dotted* lines, or rather lines composed of little strokes about $\frac{1}{16}$ in. long. (The length of these strokes varies, however, with the size of the drawing.)

3. The views of an object required are generally a side elevation, one or two end elevations, and a plan. It must be remembered, however, that the purpose of the drawing is to explain as fully as possible the object required to be made from it; it is therefore often important to show sections or imaginary cuttings through it. In this way the interior of the object can be opened up to view, thicknesses of metal shown, and the whole arrangement made much clearer to the mind of the workman. Wherever such a sectional view would help to explain the drawing, it should be given.

4. Sections are usually made through centre lines. When the sectional view is exactly similar on either side of the centre line, frequently one half view only is shown in section, and the other half in elevation. Where the plane of section passes through solid metal, that part is "cross hatched" by lines drawn at 45°. These lines might be considered as the saw marks left on the material through which the imaginary saw passed.

When two pieces of metal are in contact, and both are cut by the plane of section, the cross hatching on each is drawn in opposite directions, so as to distinguish them as separate pieces more easily. When the drawing is to be inked in, the section is indicated by a wash of colour instead of cross hatching. It is not usual to show rivets, bolts, rods, or spindles, in section. Throughout this work the method usually (though not invariably) adopted to indicate differences of material by cross hatching is as follows:—

Wrought Iron	Alternate light and heavy lines.
Cast Iron	Lines of equal thickness.
Brass	Alternate light line and dots.
Steel	Dotted lines only.

5. All the views must be so placed as to be projected the one from the other, so that their mutual connection can be readily traced. Neglect of this rule would be a fatal error.

6. Commence the drawing by putting in the principal *centre lines* first; in most cases the several parts of the object lie symmetrically on either side of this line. The centre lines should be drawn with great care, so as to be perfectly straight and square (see Direction 1); and with a sharp pointed pencil, so that measurements can be taken from it with accuracy, which is impossible with a centre line having measurable breadth.

In laying down the centre lines some care and judgment are necessary to arrange them, so that the views may be properly balanced on the paper. In the plates the centre lines have been distinguished by an alternate line and dot. The student must not imitate this, but draw a firm, fine pencil line. After the drawing has been inked in and the sections coloured, the centre lines are drawn in crimson lake.

A pencil drawing should as far as possible be correct in itself; lines which are intended to be dotted should be dotted in pencil, and not drawn in full to be dotted on inking in.

In using the pencil compasses, where many circles are to be struck from one centre, the small circles should be struck first. Care must be taken not to make large holes at the centres, as they not only look bad, but lead to inaccuracy on inking in. Needle points and double jointed instruments assist greatly to prevent this. The compass should be held upright, and not be allowed to lean over, and the needle leg bent, so that the compass may rotate on a vertical pivot.

7. Dimensions and dimension lines are not placed on a drawing which is to be inked in until after the inking in is completed, the drawing cleaned, the sections coloured, and the centre lines drawn in crimson lake. They are then put in in blue lines, a small space at or near the centre of the line being left for the dimension, which should be made in a bold, clear, black figure, such as can be easily read by a workman. The black arrow heads show the lines between which the dimension is measured. The dimensions given on a drawing indicate the full size of the *object* when constructed, no matter what the scale may be to which the drawing is made.

8. Dimensions are expressed in feet, inches, and parts of an inch, thus,—three feet, six inches, and five-eighths, is written :—$3' \cdot 6\frac{5}{8}''$. One dash indicates feet, and two dashes inches.

If the dimension required cannot be seen on one view in the plates, it may usually be found by referring to another view.

Drawing to Scale.—Wherever possible the drawing of an object is made the same size as the object itself. It is then said to be a "full size" drawing. In most cases, however, the limits of the paper prevent our representing the object full size, and we are compelled to represent it say half the full size, or 6 ins. to the foot; one-fourth the full size, or 3 ins. to the foot, and so on.

It is good practice for the student to make his own scale. To make a 3 in. scale, draw three horizontal lines about $\frac{1}{8}$ in. apart at the bottom of your paper, and mark off on the bottom line from the edge of your rule distances of 3 ins. Each of these distances is now to represent a foot. Divide the space at the left hand end into 12 equal parts to represent inches, by setting off, in this instance, quarters of an inch from the edge of the rule. We now have inches on the scale. Sub-divide one of these into halves, quarters, eighths. Several examples of scales will be found in the plates.

Now, suppose you are making a drawing from a sketch to a scale of 3 ins. = 1 ft., and a dimension on the sketch is marked 1 ft. 6 ins., this means that the *real* length of the part measures 1 ft. 6 ins., but that you are to draw it 1 ft. 6 ins. by your 3 ins. scale.

Instead of drawing from a sketch, suppose you are drawing from an actual machine: if the machine measures 8 ft. in any direction while the paper will only conveniently take a drawing 2ft. in that direction, you will make your drawing to a scale of $\frac{2}{8}$, that is $\frac{1}{4}$, or 3 ins. to the foot. If you have only 1 ft. of room your scale will be $\frac{1}{8}$ or $1\frac{1}{2}$ ins. = 1 ft., and so on. The scale should always be shown on a drawing unless that drawing is fully dimensioned.

Inking In.—The Indian ink used for inking-in drawings should be of the best quality. It is sold in sticks. The best ink has a glossy appearance, while the common is a dull, dead looking material. It is needless to say that *writing* ink should never be used, first, because its appearance on the drawing will not compare with that of good Indian ink; and secondly, because it will corrode and spoil the instruments.

To mix Indian ink, about half a teaspoonful of water should be used, and the ink rubbed in it till the liquid is somewhat thick. The ink has been rubbed sufficiently when, upon tipping the ink against the side of the saucer and allowing it to flow back again, it leaves a perfectly black mark, which retains its blackness on drying. The ink solution should always be kept covered.

When inking-in a drawing, always ink the curves and circles first, beginning with the smallest circles. It is much easier to draw straight lines to meet arcs of circles, than to draw arcs to meet straight lines.

The quality of a drawing is assessed by the clearness and firmness of the lines, and the neatness or otherwise with which the joins of straight lines with curves, or of curves with curves, are made. In a good drawing the junction of the two cannot be detected.

To ink-in a drawing well, the pens must be in good condition. Rather than waste time and patience, and eventually spoil the drawing with a bad pen, it will be much better to take the pen to the instrument makers, who will re-set it for a few pence; or the student may do it himself, as follows :—

To Re-set a Drawing Pen.—Screw up the nibs till they just touch at the points. Rub the points gently on an oil-stone by moving the pen in a plane perpendicular to the stone. Remember that the object of this is to get the two nibs exactly the same length and shape. Of course this is done at the risk of blunting the points. When both nibs are made of the same length and nicely rounded at the points, the next operation is to sharpen each nib separately on the oil-stone. To do this, open the pen and rub the nibs on their outside edges only. The wire edge formed on the inside of the nibs may be removed by rubbing gently on a piece of the finest emery paper.

Shade Lines.—Shade lines are not often used on working drawings for the workshop; but for

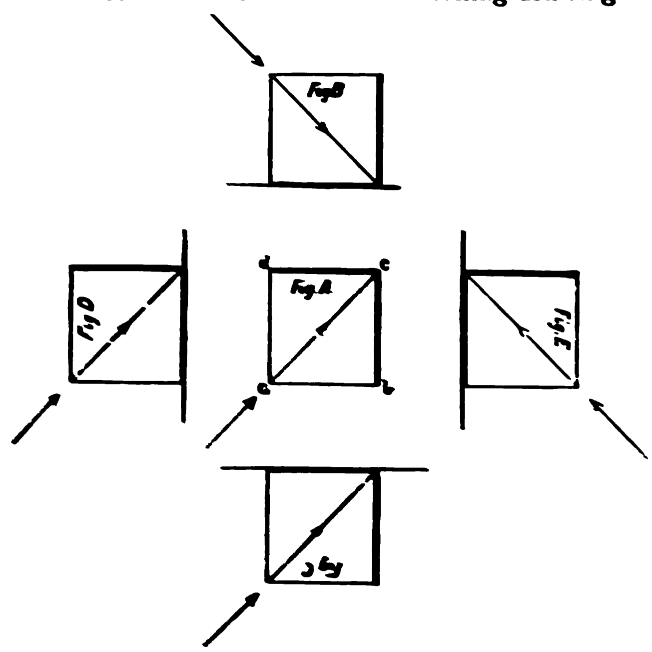


Fig. 3.

drawings prepared for a special purpose, and intended to be as well finished as possible, shade lines are added to improve their appearance. In shade lining a drawing English draughtsmen

usually follow a conventional rule, namely, that the light is supposed to proceed from the direction of the left hand top corner of the paper in parallel rays at 45° with the edge of the paper; and the shade lines on elevation, plans or sections are all drawn by this rule. This method of shade lining is admissible only on the assumption that all the views are on one plane, each view being obtained by changing the position of the *object*, instead of the position of the *plane* of projection. Thus, referring to the drawing of the loose headstock (Plate XLIX.), the sectional elevation may be spoken of as the headstock *laid on its side;* a common method of speech among draughtsmen who have not studied solid geometry. But when the different views of the object are considered as projections on different planes, the shade lines should be placed as shown by the diagram (Fig. 3).

The rays of light are considered to fall on the object in a direction parallel to one of the diagonals of a cube standing on the horizontal plane, with one face parallel to the vertical plane. The plan and elevation of this line make 45° with the ground line.

Fig. A is the plan of the cube and *a c* is the plan of the diagonal to which the rays of light are assumed parallel. Figs. B C D E are elevations of the cube on vertical planes parallel to the faces. The elevation of the diagonal in each case gives the direction of the rays and the positions of the shade lines. Shade lines should not be placed at the junction of two surfaces when both of these surfaces are visible, as in the case of the faces of a hexagonal nut. Authorities condemn the use of shade lines on the outline of a curved surface; but the practice is very common even among good draughtsmen. Shade lines should be placed so that their breadth is outside the outline of the object.

THE COLOURS USED FOR DIFFERENT MATERIALS.

MATERIAL.	COLOUR OF SECTION.
Wrought Iron	Prussian blue.
Cast Iron	Neutral tint, or Payne's grey
Steel	Purple, made by a mixture of Prussian blue and crimson lake.
Brass	Gamboge, with a touch of crimson lake and burnt umber added.
Copper	Crimson lake, softened with gamboge.
Lead	Pale Indian ink, with a little blue.
Brickwork	Light red, with a little carmine.
Wood	Yellow ochre, grained with burnt sienna.

Many a good drawing has been spoiled in the colouring. Colours either "make or mar" a drawing, and with beginners they usually have the latter effect.

TO COLOUR A WORKING DRAWING.—The student must be able to lay on a flat wash of colour, so that when dry it shall show no smears but shall look perfectly flat and even. Mix the colour in a clean saucer. On no account have it too dark. Try its tint first on a piece of waste paper. Use a moderately large brush with a fine point. Have sufficient colour in it to flow freely, though not so much as to leave pools. It is best to work downwards or from left to right. Finish all the corners as you go on, and carry the colour along quickly, on no account allowing the forward edge to dry. Do not leave any patches bare of colour, requiring you to go back over it to put in touches here and there. On finishing off the last corner of the wash, remove any little pool of colour with

the brush after emptying it first by touching it on blotting paper. The wash when dry will appear even and flat. The difficulty which beginners experience in obtaining a flat surface arises from the fact that they allow a ragged edge of the colour to *dry*, and then go over it again, thus laying on a double coat of colour in some places and only a single one in others. In colouring sections it is considered an improvement to leave a narrow strip of white margin on the edges which are towards the light.

When the colouring of sections is completed, the centre lines, which are of crimson lake, are carefully ruled in, in firm, clear lines (not dotted). It is necessary to put these lines in *after* the colouring of the sectional parts, otherwise the crimson lake would "run" over the coloured section. Lastly, the dimension lines are put in, in Prussian blue (not dotted), a space being left for the insertion of the dimension, which is written in bold, carefully-made figures with a writing pen and Indian ink. The arrow-heads are also made with a writing pen and Indian ink.

PRINTING.—It has been said that colour will either "make or mar" a drawing. This is equally true of printing, writing, and placing dimensions on a drawing.

Let the letters and figures be plain and bold. Avoid all flourishes and ornamental printing or writing. A simple though effective style of letter which can be done quickly is the best for engineering drawing.

It will be found convenient in making a heading to draw the letters first somewhat roughly on a separate slip of tracing paper so as to find out how much room they require. They may then be placed in the exact position. The following styles are suitable :—

A B C D E F G H I J K L M N O P Q R S
T U V W X Y Z
1 2 3 4 5 6 7 8 9 0

A B C D E F G H I J K L M N O P Q R S T U V W X Y Z
a b c d e f g h i j k l m n o p q r s t u v w x y z
1 2 3 4 5 6 7 8 9 0

A B C D E F G H I J K L M N O P
Q R S T U V W X Y Z

a b c d e f g h i j k l m n o p q r s t
u v w x y z

1 2 3 4 5 6 7 8 9 0

4. ON SHADING AND COLOURING FINISHED DRAWINGS.

(Plate XXXI.)

Shadows are caused when solid bodies intercept the rays of light. The parts of the solid exposed to the rays are said to be in *light* and the remainder in *shadow*. Lights and shadows vary in intensity according to the following rules :—

Rule 1.—Illuminated flat surfaces, parallel to the plane of projection, receive a uniform light tint. (See faces *a b*, *c d*, Plate XXXI., Figs. 2 and 3). If there are two or more of such surfaces at different distances from the eye, the nearest surface receives the lightest tint.

Rule 2.—Illuminated flat surfaces, inclined to the plane of projection, have a light tint, which becomes gradually darker as the surface recedes. (See faces *e a f c* of Figs. 2 and 3.)

Rule 3.—Flat surfaces in the shade, parallel to the plane of projection, receive a uniform dark tint. If there are two or more of such surfaces at different distances from the eye the nearest surface receives the darkest tint.

Rule 4.—Flat surfaces in the shade, inclined to the plane of projection, have a dark tint which becomes gradually lighter as the surface recedes. (See face *b g d h*, Figs. 2 and 3.)

These effects can be produced by two methods. (1.) By laying on a succession of light tints each overlapping the other until a proper graduation of shadow is obtained ; or (2) by proceeding as before but softening the edges of the tints as they are laid on. The first is known as the French method, and may often be made very effective, especially for large diagrams. The second requires more skill with the brush, and gives a much smoother and more pleasing effect.

To Shade by a Succession of Flat Tints.—The colour used for practice may be either Indian ink or neutral tint. Prepare three different tints of the colour, one rather dark, one pale, and one intermediate, and have two clean sable hair brushes, (not too small) and a vessel of clean water. Take first the hexagonal prism (Figs. 1, 2, 3, Plate XXXI.) ; draw the plan (Fig. 1) ; then assuming the rays of light to fall as at *b*, Fig. 1., the faces *a b*, *a e* are illuminated, *a b* being parallel to the plans of projection, and *a e* inclined to it. Also *b g* is in shadow and receding. By the above rules the tints are placed as shown in Figs. 2 and 3. The face *a b c d*, Fig. 2, has a uniform flat tint. The face *b g d h* is shaded thus :—Divide *b g* into any number of equal parts, say three, $b\ 1'$, $1'2'$, $2'g$, and draw very faint pencil lines through $1'$ and $2'$. Lay a wash of the dark tint on $b'd'1'1'$, and when this is dry a wash of the medium tint from $b'd'$ to $2'2'$. Again, when dry lay a wash of the paler tint from $b'd'$ to $g'h'$, and lastly, a wash of the medium tint over the whole face $b'd'h'g'$. The face $e'f'a'c'$ is treated similarly, but the tints used are much lighter and the darker tints are towards $e'f'$.

By varying the breadths of the divisions as the successive layers are repeated, the graduation of the shadow can be rendered very effective.

To Shade the Cylinder by the method of Flat Tints.—Plate XXXI. Draw the plan, Fig. 4. Find the line of separation of light and shadow by drawing the ray at 45°, touching the circular plan in *g*. Then the arc *g b* is in shadow, and the arc *a g* in light; and the line $g'g'$ in elevation, projected from *g* in the plan, is the line of deepest shadow. Divide the arc *f b* into say seven equal parts; then *g* 1 and *g* 2 are equal on each side of *g*, and the space between these lines

receives the darkest tint. Project lines from each division in the plan to the elevation. Again, to find the portion of the face of the cylinder which will appear most affected by the light, draw the ray of light at an angle of 45° towards the centre of the plan *c*, and, cutting the cylinder in *e*; bisect the arc *ef* in *s*; then *es* is the portion of the face which will appear most affected by the light. Project lines from these points to the elevation. The space between *e′* and *s′* is left quite uncoloured in the process of shading. Having prepared three tints of the colour required, commence by laying on the deepest tint between lines 1′ 1′ and 2′ 2′ about *g*; when this is dry, overlap this tint on both sides by another wash, and so on as before described. Use the paler tints as you proceed towards the lighter parts.

To Shade by Softened Tints.—Plate XXXI. Fig. 3 illustrates the application of this method of shading to a hexagonal prism. The position of the lights and shadows are the same as in Fig. 2. To produce the softened effect, prepare as before three shades of colour, dark, medium and pale, a vessel of clean water, and two clean brushes. To begin with let the brush be fairly full of colour, not so full that it has not proper command of the colour, and not so little that the colour will almost immediately dry. Lay on a narrow strip of dark tint along the edge *b′ d′*; *while this is still wet*, dip the point of the colour brush in clean water, thus diluting the tint in the brush, and lay on a second wash by the side of the first, rubbing into the edge of the first wash a little as you proceed along the strip. Again dip the point in water and lay on a third strip, and so on. Lastly, with a *clean* brush dipped in clean water finish off the edge. When these tints are *dry*, the process may be repeated on the same surface till any desired depth of tone is attained. On no account should the tint be touched till it is quite dry. For *large* surfaces, after applying the dark wash in the position of deepest shade, and while the edge is still wet, lay on a second wash of the medium tint about twice as wide as the first wash and slightly overlapping it. Again extend the width of shade by a third strip of the paler tint applied against the wet edge of the second, and wider than the second; finally soften the last edge with another brush dipped in clean water.

A cylinder shaded by softened tints is illustrated in Fig. 6. The position of the lights and shadows are obtained from Fig. 4 exactly as in Fig. 6. The first strip of tint is laid on about a line projected from *g*, Fig. 4, and the edge is softened on each side of it by the methods just explained. Quickness of manipulation with the brushes is necessary, and beginners are recommended to practise with pale tints until they have acquired some degree of proficiency. It should be remembered that in order to preserve a fresh, bright appearance in the colouring, the fewer washes used in order to produce the required effect the better.

When the shading appears somewhat irregular and uneven, as it will inevitably do in the work of beginners, the appearance may be improved by *stippling*; that is, by touching the surface in the light places with the point of the brush, containing a depth of colour to suit the tint required, till the parts are brought up to the right shade by the little points of colour; afterwards the whole surface should receive an even wash of colour, softened at the edge as before.

To produce an effective shaded drawing there is nothing more important than the proper placing of the light tints and dark shadows, and the draughtsman will do well to observe, that in all highly finished engravings or drawings a strict and even exaggerated use is made of the Rules 1, 2, 3, and 4, page 9, namely, that the parts near the eye are in high light and deep shade,—the virtue being in the striking *contrasts* of light and shade in the *front* of the picture; while those parts which are in the background have less contrast of light and shade in proportion as they recede from

the eye, the parts in light being less bright, and those in shade less dark. The effect of distance may be still further secured by a pale wash of Indian ink over the parts in the back of the picture, the effect of which is to tone down the lights and to give to them the appearance of remoteness.

A narrow margin of light left along the edges of surfaces in the shade adds very greatly to the distinctness and effectiveness of a drawing, the outline being distinguished by a white line on the shaded background. To leave this narrow margin and to make it perfectly even in breadth, the ruling pen may be filled with colour, and a line of colour ruled parallel to the edge and a little distance from it; while the line is still wet the brush is passed along its inner edge, and the remainder of the surface completed.

CAST SHADOWS are shadows cast by one body upon another, or upon the ground. They are often very effective in showing the true shape of the part. The correct delineation of cast shadows is explained in works on solid geometry.

5. ON HAND-SKETCHING.

No accomplishment is of more importance to the engineer than the power to make a hand-sketch from a machine as a whole or in detail. Drawing from actual machine details is, however, by no means a first step in mechanical drawing, but should be commenced when the student has become acquainted with the methods of representation by plan, section, and elevation, the proper placing of dimensions, &c.

The following tools are necessary:—Sketch-book, pencil, bow-pencil compass, callipers (internal and external), two-foot rule, and, if for large work, a long wooden staff divided to eighths of an inch.

The hand-sketches of details should be drawn as nearly as possible after the manner of those shown in the various plates in this work.

If the drawing of a complete machine is required, a freehand sketch of the general outline, omitting small details, is first made, special care being taken to get the positions of the principal centres relatively to one another or to two fixed lines at right angles, the vertical heights being measured from a base or foundation line, and the horizontal distances from some line through a main centre at right angles to the base line. Working links or levers may be shown in position merely by their centre lines, which may be lettered or numbered and then sketched in detail on a separate page to a scale large enough to show all the necessary dimensions.

A sketch should on no account be overcrowded with details.

To obtain the dimensions between the centres of holes *c* and *d*, Fig. 4, measure from edge *a* to edge *b*; if the holes are unequal, measure the distance *a e* and add to it half the diameter of each hole. If the circles represent shafts, measure the distance *a e* between the shafts and add to it half the diameter of each shaft.

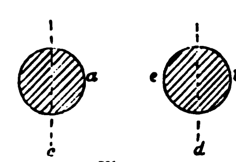


Fig. 4.

The diameters of toothed wheels are given to the pitch lines (see Plate XLIII.), and the pitch of the teeth is measured on the pitch line.

3

Chapter II.

6. MENSURATION OF SURFACES AND SOLIDS.

1. Area of rectangle .. ... = length × breadth.
2. Area of triangle ... ... = base × half perpendicular height.
3. Diameter of circle ... ... = radius × 2.
4. Circumference of circle ... ... = diameter × 3·1416.
5. Area of circle ... ... = diameter × diameter × ·7854.
6. Area of sector of circle ... ... = $\frac{\text{area of circle} \times \text{No. of degrees in arc.}}{360.}$

7. To find the diameter of a circle, having given the area: Divide the area by ·7854, and extract the square root.

8. To find the solid contents of a bar of uniform cross section: Multiply the area of the section in square inches by the length of the bar in inches.

9. To find the solid contents of a sphere: Multiply the cube of the diameter by ·5236.

Ex. 1. Find the number of square feet of sheet lead (neglecting laps) to completely line the inside of a tank, open on the top, 6 ft. long, 4 ft. wide, 3 ft. deep. *Ans.* 84 square feet.

Ex. 2. What weight of fresh water will fill the tank? 1 cubic foot = 62·5 lbs. *Ans.* 4,500 lbs.

Ex. 3. Find the heating surface of a cylindrical furnace flue, 3 ft. diameter and 6 ft. long (one half the surface is effective). *Ans.* 28·2744 square feet.

Ex. 4. A cast-iron pipe is 6 ins. internal diameter, and the thickness of metal is 1 inch; find the area of metal in a transverse section of the pipe. *Ans.* 21·99 square ins.

TABLE I. WEIGHT OF METALS.

	Per cubic inch.	Per cubic foot.
Wrought-iron	·28 lbs.	480
Cast-iron	·26 ,,	450
Steel (soft, untempered) ...	·28 ,,	480
Steel (cast)	·29 ,,	496
Brass	·30 ,,	520
Copper	·32 ,,	550

1728 cubic inches = 1 cubic foot.

Ex. 1. Find the weight of a piece of steel shafting, 3 ins. diameter and 10 ft. long.
Ans. $237\frac{1}{2}$ lbs., nearly.

Ex. 2. A safety valve is loaded with 7 cast-iron weights, 10 ins. diameter and $1\frac{1}{2}$ ins. thick; find the weight on the valve (neglecting the hole through the weights, which is filled by the valve spindle). *Ans.* 214·4142 lbs.

Ex. 3. The safety valve, loaded as in above example, is $2\frac{1}{2}$ ins. diameter; find the pressure per square inch on the valve. *Ans.* $43\frac{2}{3}$ lbs. per square inch.

Ex. 4. A cast steel cylinder liner is 42 ins. inside diameter, 4 ft. 6 ins. long, and $1\frac{1}{4}$ ins. thick; find its weight. *Ans.* 2662 lbs.

7. STRENGTH OF MATERIALS. PRINCIPLES OF CONSTRUCTION.

The strength of materials is determined by the resistance which they offer to change of form, and ultimately to fracture, when subjected to the action of various kinds of forces.

The internal molecular resistance which a body offers to change of form, and which is equal and opposite to the acting force, is called the *stress* in the body.

The term *stress*, however, is used indifferently to express either the acting force or the internal resistance. Stresses are expressed in lbs. or tons per square inch of sectional area.

The *change of form* produced in a body by the acting force is called the *strain*. Strains are measured, not in lbs. or tons, but in inches, or parts of an inch.

Stresses are :—

Tensile, tending to cause fracture by pulling asunder.
Compressive ... ... crushing.
Transverse ... ... breaking across.
Shearing ... ... cutting across.
Torsional ... ... twisting asunder.

The *ultimate strength* of a body is the greatest load required to produce fracture, and it is expressed in tons or lbs. per square inch of cross sectional area.

The *working strength* is the load which the material will safely bear under the given conditions, after allowing a sufficient margin for safety. The working strength is some fraction of the ultimate strength which is determined by experience.

The mean values of the strength of materials on the following page are given by Professor W. C. Unwin in his work on " Machine Design."

TABLE II. ULTIMATE AND ELASTIC STRENGTH OF MATERIALS IN POUNDS PER SQUARE INCH.

Material.	Breaking Strength.			Elastic Strength.		
	Tension.	Compression.	Shearing.	Tension.	Compression.	Shearing.
Cast-iron	17,500	95,000	28,500	10,500	21,000	7,900
Wrought-iron bars ...	57,600	50,000	50,000	24,000	24,000	20,000
Wrought-iron plates—						
with fibre ...	50,700					
across fibre ...	46,100					
mean	48,400	...	...	20,000	20,000	15,000
Steel boiler plates ...	66,000	...	...	31,000		
Rivet steel	65,000	...	55,600			
Soft steel, unhardened	80,000	...	...	35,000	...	26,500
Cast steel, untempered	120,000	...	...	80,000	...	64,000
Copper	33,000	58,000	...	4,300	3,900	2,900
Brass, yellow	17,500	10,500	...	6,950	...	5,200
Gun metal	36,000	...	...	6,200	...	4,150
Phosphor bronze ...	58,000	...	...	19,700	...	14,500

TABLE III. ORDINARY WORKING STRESS.

Material.	Safe Limit of Stress in lbs. per sq. in.		
	Tension.	Compression.	Shearing.
Cast-iron	3,600	10,400	2.700
Wrought-iron bars	10,400	10,400	7,800
Wrought-iron plates	10,000	10,000	7,800
Soft steel, untempered ...	17,700	17,700	13,000
Cast steel, untempered ...	52,000	52,000	38,500
Copper	3,600	3,120	2,300
Brass	3,600	..	2,700
Gun metal	3,120	...	2,400
Phosphor bronze	9,870	...	7,380

The *Factor of Safety* is the ratio of the ultimate strength to the working strength. Thus, if the ultimate strength per sq. in. of wrought-iron be 50,000 lbs., and the working strength 10,000 lbs.,

$$\text{Factor of Safety} = \frac{\text{ultimate strength}}{\text{working strength}} = \frac{50{,}000}{10{,}000} = 5$$

The factor of safety chosen for materials under the action of moving, or "live" loads, is much greater than when under the action of steady or "dead" loads. The following table gives the minimum values of this factor :—

TABLE IV. FACTOR OF SAFETY.

MATERIAL.	Dead load.	Live load.	When subject to shocks.
Wrought-iron and steel ...	3	5	10
Cast-iron	4	6	10

The *proof or testing strength* is the greatest load which, on continued application, does not injure the material.

The breaking strength of a bar of cast-iron subjected to a tensile load can be determined with exactness; but with a tough, ductile material, such as wrought-iron or steel, it is not easy to determine so exactly the breaking point. When a wrought-iron or steel bar is placed in a testing machine and a tensile load considerably less than the breaking load is applied to it, it may be observed by careful measurement that the bar has been stretched. During the earlier part of the test, as the load is gradually increased, the amount of the extension or strain is found to be proportional to the load or stress applied. If the load be removed at any time during this stage, the bar will recover its original form almost exactly, and so far the bar is perfectly elastic. Any small deformation of the bar is called *permanent set.* When, however, the load approaches the breaking weight, the proportionality previously observed between stresses and strains now ceases, and the extensions rapidly increase in proportion to the load, and this continues until the maximum load is added which the bar will bear, when it breaks.

The *elastic limit* is the point at which the proportionality between stresses and strains ceases when a bar is being subjected to a gradually increasing load.

TENSILE AND COMPRESSIVE STRESS.—Simple tensile or compressive stresses acting on a bar are regarded as being evenly distributed over the cross sectional area of the bar, so that the stress per square inch = load ÷ area; or, if P = the load carried by the bar, a = the area of the bar in sq. ins., and f = the stress per sq. in., then

$$f = \frac{P}{a} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (1)$$

Ex. 1. What is the stress per sq. in. on a bar 4 ins. diameter subjected to a tensile load of 25 tons?

$$\text{Stress per sq. in.} = \frac{\text{load}}{\text{area}}$$

$$= \frac{2240 \times 25}{4 \times 4 \times \cdot 7854} \qquad \begin{array}{l}(2240 \text{ lbs.} = 1 \text{ ton})\\(d^2 \times \cdot 7854 = \text{area})\end{array}$$

$$= 4456 \cdot 328 \text{ lbs. } \textit{Answer.}$$

Ex. 2. Find the tensile load which can be carried by a bar 1·25 in. diameter, allowing a safe working stress of 10,000 lbs. per sq. in.

$$\text{From } f = \frac{P}{a}$$

$$\begin{aligned} \text{We have } P &= f\,a \\ \text{or, load} &= \text{stress} \times \text{area} \\ &= 10{,}000 \times 1{\cdot}25 \times 1{\cdot}25 \times {\cdot}7854 \\ &= 20126 \text{ lbs.} \\ &= 9 \text{ tons, nearly.} \end{aligned}$$

Ex. 3. What should be the area of a bar to carry a load of 50,000 lbs., allowing a safe stress of 5,000 lbs. per square inch?

$$\begin{aligned} a &= \frac{P}{f} \\ \text{or, area} &= \frac{\text{load}}{\text{safe stress}} \\ &= \frac{50000}{5000} \\ &= 10 \text{ sq. ins.} \end{aligned}$$

Ex. 4. Find the diameter of a round bar with a sectional area of 10 sq. ins.

$$\begin{aligned} \text{Area} &= \text{diameter}^2 \times {\cdot}7854 \\ \text{or, diameter}^2 &= \frac{\text{area}}{{\cdot}7854} \\ \text{diameter} &= \sqrt{\frac{10}{{\cdot}7854}} \\ &= 3{\cdot}57 \text{ ins.} \end{aligned}$$

Ex. 5. If the crushing strength of a sample of cast-iron is 40 tons per square inch, what weight will crush a short pillar 6 ins. diameter?

$$\begin{aligned} \text{Crushing load} &= \text{strength per sq. in.} \times \text{area} \\ &= 40 \times 6 \times 6 \times {\cdot}7854 \\ &= 1130{\cdot}976 \text{ tons.} \end{aligned}$$

Ex. 6. A wrought-iron bar 1 inch diameter broke with a tensile load of 45,000 lbs.; find the tensile strength of the bar per square inch. *Ans.* 57,295·645 lbs.

Ex. 7. The strength per square inch of wrought-iron bars averages 57,600 lbs.; find the breaking load of a bar 2 ins. diameter, and of a bar 3 ins. diameter, and compare the results.

Ex. 8. A manganese steel bar 1 inch diameter at smallest section broke with a tensile load of 21·83 tons; find the breaking strength of the bar per square inch. *Ans.* 27·8 tons.

Ex. 9. A cast-iron bar 0·875 inch diameter broke with a tensile load of 7·75 tons; find the breaking strength per square inch. *Ans.* 12·9 tons.

Ex. 10. A wrought-iron bar 0·875 inch diameter broke with a tensile load of 12·19 tons; find the breaking strength per square inch. *Ans.* 22 tons.

Ex. 11. A brass bar 0·625 inch diameter broke with a tensile load of 3·17 tons; find the breaking strength per square inch. *Ans.* 10·3 tons.

Ex. 12. A piece of manganese steel plate of rectangular section 2 ins. × $\frac{3}{8}$ in. broke with a tensile load of 22·935 tons; find the breaking strength per square inch. *Ans.* 30·58 tons.

SHEARING STRESS.—When a force acts on a body, and tends to cause separation of the parts at a given section by the sliding of the parts one over the other, the body is subjected to a *shearing stress*. This is illustrated by the action of a pair of scissors, or of the tool in a shearing machine, or by the way in which a rivet securing two plates may be cut across on the plane passing between the plates. Other examples of shearing are illustrated under pins, cotters, &c.

For simple cases of shearing, if a = the area of the section in sq. ins., P = the acting force, and f = the shearing stress per sq. in., then

$$f = \frac{P}{a}$$

A similar action, namely, a shearing or detrusion, of the material takes place when a hole is punched through a solid plate.

The following are particulars of a set of experiments on punching circular holes in wrought-iron plates :—

Complete the table.

Diameter of punch, in inches.	Thickness of plate, in inches.	Total weight on punch, in tons.	Sectional area of detrusion, in sq. inches.	Stress per sq. inch of area, in tons.
0·250	0·437	8·384		
0·500	0·625	26·678		
0·750	0·625	34·768		
0·875	0·875	55·000		
1·000	1·000	77·170		

Thus :—

Sectional area of detrusion = diameter × 3·1416 × thickness.

$$\text{Stress per square inch} = \frac{\text{total weight on punch.}}{\text{sectional area of detrusion.}}$$

When pins or cotters are subjected to the action of a shearing stress, it is usual to allow additional sectional area for possible looseness of fitting of the parts, as follows :—Where f = the working shearing stress per sq. in. :

$$\text{For rectangular cross sections } a = \frac{3}{2} \times \frac{P}{f}$$

$$\text{For circular cross sections } a = \frac{4}{3} \times \frac{P}{f}$$

TRANSVERSE STRESS. STRENGTH OF BEAMS SUBJECTED TO BENDING.—A bar resting upon supports at both ends is a simple *beam*. A bar secured at one end and unsupported at the other is called a *cantilever*.

A loaded beam resists fracture so long as the external forces acting upon it, namely, the loads and reactions at the supports, do not exceed the internal molecular resistances of which the beam itself is capable.

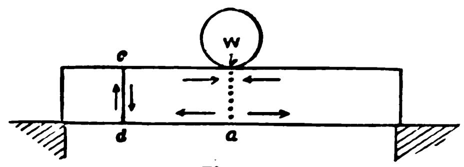


Fig. 5.

The *external forces* tending to fracture the beam may be resolved into (1) horizontal stresses, tending to tear the fibres asunder at *a* (Fig. 5) in the direction of the length of the beam, and to crush the fibres at *b*; (2) a vertical shearing stress, tending to cause fracture by shearing through any vertical section as at *c d*.

Opposed to the external forces is the resistance offered by the fibres of the beam, acting about the neutral axis of the section; and the general condition of equilibrium is, that the bending moment of the external forces = moment of the internal resistances.

The bending moment can be easily calculated, as explained below: and the moment of resistance for any given section may be obtained by multiplying the tensile strength of the material (f) by a value which depends upon the form and dimensions of the section, and known as the modulus of the section, for bending stress, as given in Table VI.

Definitions.—The *moment of a force about a point* is the product of the force into the perpendicular distance of the point from the line of action of the force.

The *bending moment* at any point in a beam is the sum of the moments about that point of all the external forces acting upon the beam on either side of the point.

Reactions at the Supports.—If a weight rest upon a beam which is supported at both ends, the *sum of the reactions* at the ends *is equal to the weight*, and the amount of the reaction at each end is to the whole weight, as the remote segment of the beam is to the whole beam. Thus, if W = the load (Fig. 6), and R and R_1 = the resistances, then, always

$$R + R_1 = W.$$

If W be in the centre of the beam, then evidently R and R_1 each equal $\frac{1}{2}$ W,

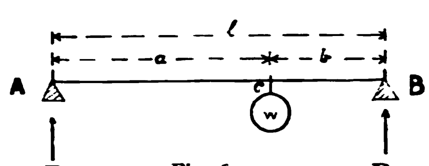


Fig. 6.

but if W act at a distance a from R then

$$R : W :: b : l$$

$$\text{Therefore reaction } R = \frac{W\,b}{l}$$

$$\text{also reaction } R_1 = \frac{W\,a}{l}$$

Ex. A beam 10 feet long, supported at both ends, carries a load of 9000 lbs. 7 feet from the end A; find the reactions R and R_1 at A and B (Fig. 6).

$$R = \frac{W\,b}{l} = \frac{9000 \times 3}{10} = 2700 \text{ lbs.}$$

$$R_1 = \frac{W\,a}{l} = \frac{9000 \times 7}{10} = 6300 \text{ lbs.}$$

(Notice 2700 + 6300 = 9000 lbs., the total weight W.)

The same result is obtained by taking the moments of forces about any point in the beam. Thus, take moments about the point B as a hinge or fulcrum; R tends to turn the beam about the hinge B in one direction, while W tends to turn it in the opposite; but the beam is in equilibrium, and when a body acted upon by forces is in equilibrium, the moments of the forces tending to turn it in one direction round a given point are equal to the moments of the forces tending to turn it in the opposite direction.

$$\therefore R \times l = W \times b$$
$$\text{or } R \times 10 = 9000 \times 3$$
$$\therefore R = 2700 \text{ lbs., resistance at support A.}$$

Similarly by taking moments about A, resistance at support B may be found. It is also obtained, of course, by subtracting the resistance at A from the total weight;

$$9000 - 2700 = 6300 \text{ lbs.}$$

The bending moment in the beam is greatest under c, the point at which the weight is supported, and it here equals $R \times a$, or $R_1 \times b$.

$$\text{But } R = \frac{W\,b}{l}$$

Therefore bending moment about $c = R \times a = \dfrac{W\,a\,b}{l}$

Ex. Find the bending moment about the point c in case given in the previous example.

$$\text{Bending moment} = \frac{W\,a\,b}{l} = \frac{9000 \times 7 \times 3}{10} = 18900$$

We will now apply these principles to the case of a simple wrought-iron flanged girder of section, shown in Fig. 7.

In a flanged girder with a thin web, the flanges are considered to bear the whole of the horizontal stresses due to the bending action of the load, and the web the vertical or shearing stresses.

Let W = load; l = length in inches; d = depth in inches (measured from the centre of gravity of the flanges); T = tensile stress in top flange; and C = compressive stress in bottom flange.

When the girder is fixed at one end and loaded at the other (Fig. 7), moment of load W tending to bend the beam about a as a hinge $= W \times l$; moment of resistance of material in top flange opposing the bending and tending to turn the beam in the opposite direction about $a = T \times d$.

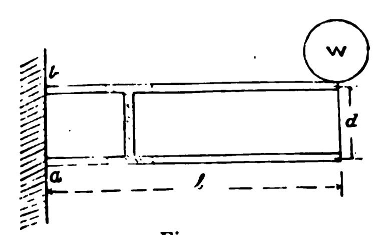


Fig. 7.

Then $T \times d = W \times l$.

Again, taking moments about b,

$$\text{Bending moment} = W \times l$$
$$\text{Moment of crushing stress in bottom flange} = C \times d$$
$$\text{And } W \times l = C \times d$$
$$\text{Therefore } T \times d = C \times d$$
$$\text{or } T = C$$

That is, tensile stress in top flange = compressive stress in bottom flange.

4

Wrought-iron beams are usually made with equal flanges, because the strength of the material to resist both tension and compression is about equal.

Ex. A flanged wrought-iron cantilever is loaded at the free end, $W = 2$ tons, $l = 6$ ft., $d = 9$ ins.; find the stresses in the flanges.

Moment of resistance = bending moment.

$$T \times d = W \times l$$
$$T \times 9 = 2 \times 72$$
$$T = 16 \text{ tons.}$$

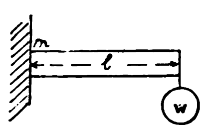
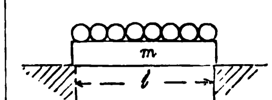
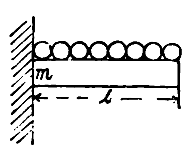
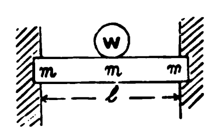
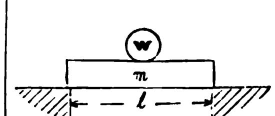
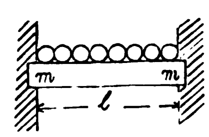
Allowing 4 tons per sq. in. safe working stress both for tension and compression, then the sectional area of each flange should be—

$$\text{Area} = \frac{\text{load}}{\text{stress}} = \frac{16}{4} = 4 \text{ square inches.}$$

The shearing stress in the web is equal to the load, namely, 2 tons; and in this case it acts equally throughout the length of the girder; and allowing 4 tons per sq. in. as for tension and compression, the sectional area of the web would require to be half a square inch. In practice the strength of the web usually exceeds that necessary to bear the shearing stress.

The *shearing stress* at any cross section of a beam is the sum of the external forces acting on either side of that section, and comprises the supporting force (if any) acting in one direction, and the loads on that side, acting in the opposite direction; the forces acting in one direction being positive, and those acting in the other being negative.

TABLE V. RESISTANCE OF BEAMS TO BENDING.

	Position of load and point of support.	Greatest bending moment at *m*.		Position of load and point of support.	Greatest bending moment at *m*.
I.		W L	IV.		$\frac{WL}{8}$
II.		$\frac{WL}{2}$	V.		$\frac{WL}{8}$
III.		$\frac{WL}{4}$	VI.		$\frac{WL}{12}$

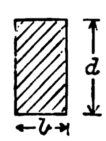
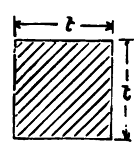
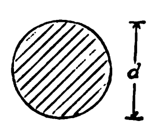
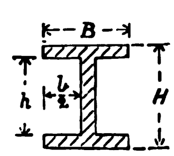
From the above table we may now compare the relative strength of beams under different conditions of load and points of support.

	Relative strength.
When supported at one end and loaded at the other	1
When supported at one end and load uniformly distributed	2
When supported at both ends and loaded at centre	4
When supported at both ends and load uniformly distributed	8
When rigidly secured at both ends and loaded in the centre	8
When rigidly secured at both ends and load uniformly distributed ...	12

RESISTANCE OF BEAMS TO TRANSVERSE OR BENDING STRESS.

In the case of the wrought-iron flanged girder, the flanges have been considered to resist the horizontal stresses, and the web the vertical stresses. But in other forms of section the stresses cannot be so definitely located, and, as before stated, the moment of resistance of a beam of any given section is found by multiplying the strength of the material to resist tension by the ***modulus of section***, the value of which for simple sections is given in the following table :—

TABLE VI. MODULUS OF SECTION (*Z*) FOR SIMPLE BEAMS SUBJECTED TO BENDING.

Form of section.	Modulus of section Z	Area of section.
	$\dfrac{b\,d^2}{6}$	$b\,d$
	$\dfrac{t^3}{6}$	t^2
	$\dfrac{d^3}{10{\cdot}2}$	$d^2 \times {\cdot}7854$
	$\dfrac{B\,H^3 - b\,h^3}{6\,H}$	$B\,H - b\,h$

Suppose a bar or beam of any section to be loaded in the centre and supported at both ends, then—

Bending moment = moment of resistance.

$$\frac{W\,l}{4} = f \times z$$

From which $W = \frac{4 f z}{l}$

Where W = breaking load, f = the breaking tensile strength per square inch, l = length of bar between supports in inches, b = breadth in inches, and d = depth in inches. This simple rule gives results which correspond with considerable accuracy to those obtained by experiment, *except for the case of the solid sections*, such as the rectangular, square, and circular sections, and for these the breaking weight W, given by the formula, is very much less than that obtained by actual experiment. But since in machine construction these simple sections are almost the only ones used, it is necessary that we should be able to estimate their strength with accuracy, and the above formula, which is a very handy one, may still be used, provided the value of f be taken from the following table, when t = ultimate tensile strength of material per square inch.

TABLE VII. VALUE OF f FOR SOLID SECTIONS.

	Cast-iron.	Wrought-iron.	Steel.
Rectangular section	$2\frac{1}{4}\,t$	$1\frac{1}{2}\,t$	$1{\cdot}7\,t$
Circular section	$2\frac{1}{2}\,t$	$1\frac{3}{4}\,t$	$1{\cdot}8\,t$

Ex. 1. Find breaking load of a bar of cast-iron 12 ins. long, 2 ins. broad, and 4 ins. deep, supported at both ends and loaded in the centre.

Bending moment = moment of resistance

$$\frac{W\,l}{4} = \frac{f\,b\,d^2}{6} \quad \text{(See Tables II., V., VI., VII.)}$$

$$W = \frac{4 \times 17500 \times 2\frac{1}{4} \times 2 \times 16}{6 \times 12}$$

$$= 70000 \text{ lbs.}$$

Ex. 2. Find the dimensions of the rectangular section of a cast-iron bar 12 ins. long which will break under a load of not less than 70,000 lbs. when supported at both ends and loaded in the centre.

As before $\frac{W\,l}{4} = \frac{f\,b\,d^2}{6}$

or $b\,d^2 = \frac{6\,W\,l}{4\,f} = \frac{6 \times 70000 \times 12}{4 \times 17500 \times 2\frac{1}{4}} = 32$

Let $b = \frac{1}{2}\,d$; then $b\,d^2 = 32$

$\frac{1}{2}\,d^3 = 32$

$d^3 = 64$

$d = 4$ ins.

and $b = 2$ ins.

Ex. 3. A wrought-iron bar $3\frac{1}{2}$ ins. diameter and 2 feet long is supported at both ends and carries an evenly distributed load; what will be the extent of this load which will strain the bar to its elastic limit?

(From Table II. f for elastic strength of bar = 24,000 lbs.)

$$\frac{W\,l}{8} = \frac{f\,d^3}{10{\cdot}2}$$

$$W = \frac{8\,f\,d^3}{10{\cdot}2\,l}$$

$$= \frac{8 \times 24000 \times 1\frac{5}{8} \times 3\frac{1}{2} \times 3\frac{1}{2} \times 3\frac{1}{2}}{10{\cdot}2 \times 12 \times 2}$$

$$= 54644 \text{ lbs.}$$

Ex. 4. A steel cotter 4 ins. thick by 9 ins. deep, and 9 ins. long between points of support, is subjected to an evenly distributed load tending to bend the cotter; find the extent of the load which would strain the cotter to its elastic limit.

$$\frac{W\,l}{8} = \frac{f\,b\,d^2}{6}$$

$$W = \frac{8\,f\,b\,d^2}{6\,l}$$

$$= \frac{8 \times 35000 \times 1{\cdot}7 \times 4 \times 81}{6 \times 9}$$

$$= 2856000 \text{ lbs.}$$

$$= 1275 \text{ tons.}$$

Tension or Compression combined with Bending.—When a bar B (Fig. 8) is loaded in such a way that the line of action of the load does not coincide with the axis of the bar, the effect on any section ($a\,c$) of the bar is equivalent to that due to a tensile load acting directly through the axis, together with a bending action, about the neutral axis (n) of the section, whose moment is equal to $W\,l$.

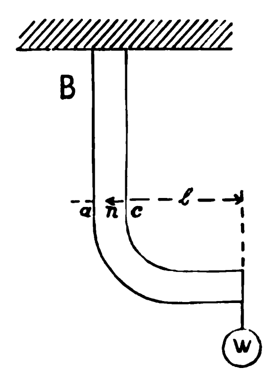


Fig. 8.

The stress per square inch (f) in the bar due to tension $= \frac{W}{a}$ ... (1)
when W = the load, and a = area of cross section of the bar.

The stress due to the bending action is obtained from the following equation:—

Moment of resistance = bending moment.

$$f\,z = W\,l \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

where z = the modulus of the section of the bar (p. 21).

But the strength of the material to resist bending is reduced by the stress $\frac{W}{a}$, due to accompanying tension, and the equation (2) now becomes

$$\left(f - \frac{W}{a}\right) z = W\,l \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (3)$$

Applying this formula to the solution of the stresses in a crane hook (Fig. 9), for example, the stress is greatest at the section $a\,b$, at right angles to the line of action of the load, where the bending moment, $W \times l$, is greatest.

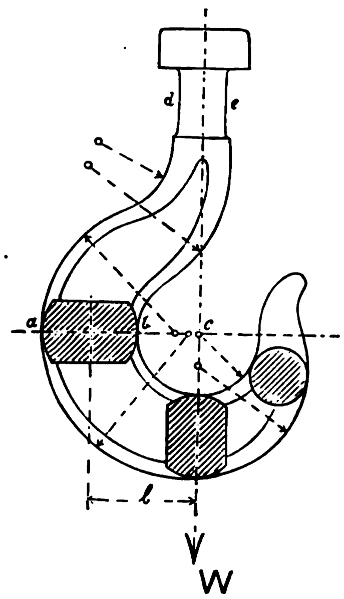


Fig. 9.

Let the cross section at $a\ b$ be equivalent to a rectangular section $3\frac{1}{4}$ ins. by 2 ins., and let $l = 4$ ins., z for rectangular sections $= \frac{b\ d^2}{6}$; find the breaking load W.

$$\text{Then } \left(f - \frac{W}{a}\right) z = W\ l,$$

$$\text{or, } \left(f - \frac{W}{b\ d}\right) \frac{b\ d^2}{6} = W\ l$$

but by Table VII., page 22, the value of f, the ultimate tensile strength, may be multiplied by $\frac{3}{2}$ for wrought-iron subjected to bending, when we have

$$\frac{3}{2}\left(f - \frac{W}{b\ d}\right) \frac{b\ d^2}{6} = W\ l, \text{ from which}$$

$$W = \frac{\frac{3}{2} f\ b\ d^2}{6\ l + d}$$

Substituting the values given above for the dimensions of the hook, and taking f, the ultimate tensile strength of wrought-iron, $= 25$ tons per square inch ; we can now find the breaking load.

$$W = \frac{1{\cdot}5 \times 25 \times 2 \times 3{\cdot}25 \times 3{\cdot}25}{6 \times 4 + 3{\cdot}25}$$

$$= 29 \text{ tons, nearly.}$$

It is, of course, necessary that the area of the circular section through $d\ e$, the weakest part, should be at least sufficient to bear a tensile load equal to the above.

Torsion or Twisting.—If one end of a horizontal bar be rigidly secured, and if to the free end a lever be attached at right angles to the bar, and a weight be suspended from the end of the lever, then the bar will be subjected to a twisting force, and its strength to resist twisting will depend upon the length of the lever and the amount of the weight. The weight multiplied by the perpendicular distance between its line of action and the axis of the bar is called the *twisting moment.* The stresses which occur in a bar under a twisting force are really shearing stresses, for the tendency is evidently to cause one part of the bar to rotate relatively to the part adjacent to it.

Opposed to the external twisting action is the internal resistance of the material of the bar, and the condition of equilibrium is that the twisting moment = the moment of resistance. If P = the load on the lever, and l = the length of the lever from centre of line of action of load to the axis, then P l = twisting moment.

The moment of resistance to twisting of any section is proportional to the shearing strength of the material, multiplied by a value depending on the form and dimensions of the section, and known as the modulus of the section for torsion.

TABLE VIII. MODULI OF SECTIONS FOR TORSIONAL STRESS.

Form of section.	Modulus.
Solid circular section ... diameter $= d$	$\frac{\pi}{16} d^3 = \frac{d^3}{5 \cdot 1}$
Hollow cylindrical section ... $d_1 =$ outside diameter $d_2 =$ inside diameter	$\frac{\pi}{16} \times \frac{d_1^4 - d_2^4}{d_1}$

The formula for obtaining the diameter (d) of a shaft, subjected to twisting stress, may be obtained from the general formula, thus:

Moment of resistance = twisting moment.

$$f \frac{d^3}{5 \cdot 1} = P\,l$$

$$d^3 = \frac{P\,l}{f} \times 5 \cdot 1$$

$$d = \sqrt[3]{\frac{P\,l}{f} \times 5 \cdot 1}$$

where $f = 8{,}000$ for wrought-iron and 12,000 for steel.

Ex. 1. Find the diameter of a solid wrought-iron shaft to transmit 100 horse power at 150 revolutions per minute.

For each revolution we have $\frac{100 \times 33000}{150} = 22000$ ft. lbs. of work performed.

This is equivalent to what would be performed by a weight equal to $\frac{22000}{3 \cdot 1416 \times 2} = 3500$ lbs. acting on a wheel 1 ft. radius

Therefore the twisting moment $= 3500 \times 1 = 3500$ ft. lbs., or $3500 \times 12 = 42000$ inch lbs.

$$d = \sqrt[3]{\frac{\text{twisting moment}}{f} \times 5 \cdot 1}$$

$$= \sqrt[3]{\frac{42000}{8000} \times 5 \cdot 1}$$

$$= \sqrt[3]{26 \cdot 775}$$

$= 3$ ins. nearly. (By calculation, or by table of cube roots.)

Ex. 2. If a wrought-iron shaft 1 inch diameter break by torsion with a load of 800 lbs. acting at the end of a 12 inch lever, or at the circumference of a wheel 2 ft. in diameter, what load would break the shaft if it were increased to 3 ins. diameter?

Since the strength of shafts is in proportion to the *cubes* of their diameters,

Then $1^3 : 3^3 :: 800 : x$

$$x = \frac{800 \times 27}{1} = 21600 \text{ lbs.}$$

Hollow Shafts.—Hollow steel crank shafts are now frequently used, especially in marine engines, where economy in weight is of the greatest importance. The diameter of a hollow shaft, to be of equal strength to that of a solid shaft, is found by equating their moduli of section with respect to torsion (see page 25) thus:

$$\frac{d_1^4 - d_2^4}{d_1} = d^3$$

Then if d_2, the internal diameter, be made equal to, say, 0·4 d_1 (d_1 being the external diameter), the equivalent hollow shaft can be obtained as follows:—

$$\frac{d_1^4 - (0{\cdot}4\,d_1)^4}{d_1} = d^3$$

$$\cdot 9744\, d_1^3 = d^3$$

$$d_1 = 1{\cdot}01 \text{ dia}^r\text{. of solid shaft.}$$

Ex. 3. Compare the resistance to torsion of two crank shafts each 10 ins. diameter; one shaft is solid, and the other is hollow, having a 4 ins. hole through it from end to end.

Let d = diameter of solid shaft, and d_1 and d_2 = outside and inside diameters of hollow shaft; then

$$\text{Solid shaft : hollow shaft} :: d^3 : \frac{d_1^4 - d_2^4}{d_1}$$

$$\text{or, as } 10^3 : \frac{10^4 - 4^4}{10}$$

$$1000 : 974{\cdot}4$$

indicating a loss of only 2·56 per cent. in strength to resist torsion for a reduction of 16 per cent. in weight.

Combined Twisting and Bending.—The application of a twisting stress to a shaft usually gives rise to an accompanying bending action about the point or points of support of the shaft.

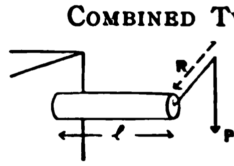


Fig. 10.

These combined stresses may be expressed as a simple bending moment equivalent to them, and called the *equivalent bending moment* M^1, thus:—

$$M^1 = \tfrac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$$

or, more simply,

$$M^1 = \cdot 9\,M + \cdot 4\,T \text{ (approximately)},$$

where M = the bending moment $P\,l$, and T = the twisting moment $P\,r$.

The *equivalent twisting moment* T^1 is given by the following equation:—

$$T^1 = M + \sqrt{M^2 + T^2}$$

Chapter III.

8. MATERIALS OF CONSTRUCTION.

CAST-IRON is composed of pure iron with from 3 to 5 per cent. of carbon, which is more or less completely combined with the iron. The varieties of cast-iron depend upon the proportion of the carbon in actual combination with the iron. When the combination is complete, it is called ***white cast-iron***, from the appearance of the fracture. This material is excessively hard and brittle, and by itself is unsuitable for castings. When a large proportion of the carbon is present in a separate form, being merely mixed with the iron, it gives the fracture a grey appearance, and it is then called ***grey cast-iron***. The grey irons are not so strong as the white, but are softer and tougher, and when mixed with the white in suitable proportions, form the ordinary foundry mixtures for castings. Cast-iron is crystalline in fracture and not nearly so strong as wrought-iron under a tensile stress, but stronger than wrought-iron under a crushing stress. Hence it may be used for columns to support weight, but not for tie rods from which weights are suspended. The chief difficulty in its use arises from the internal stresses set up in the casting owing to unequal contraction on cooling, and which are sometimes of sufficient intensity to cause fracture before any load has been applied. This danger may be prevented to a great extent by making the thickness of the parts as uniform as possible, and by avoiding sharp corners. The reason of this will be obvious on examining the following diagrams. The lines on the figures represent the grouping of the crystals of the metal,

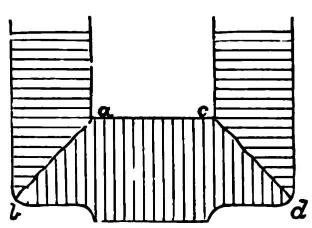


Fig. 11A.

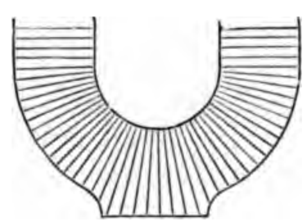


Fig. 11B.

the principal axes of which are said to arrange themselves perpendicularly to the cooling surface of the solid. In Fig. 11A, the arrangement of the crystals shows a plane of weakness to exist along the lines *ab* and *cd*; while in Fig. 11B, no such plane of weakness exists. Fig. 11A represents the lower end of the cylinder of the hydraulic press as at first made to lift the tubes of the Britannia Bridge into position, and which broke in the attempt, the fracture taking place all round the bottom of the cylinder at the sharp corner. A new cylinder was made no thicker than before but of the form shown in Fig. 11B. This stood the test without injury. The average strength of cast-iron to resist crushing is 42 tons per square inch, and to resist a tensile stress 7 tons per square inch, or one-sixth its strength to resist crushing.

WROUGHT-IRON is a fibrous, tough, ductile, malleable material. It is made from cast-iron by abstracting the greater part of the carbon, and it is manufactured in the form of bars or plates. It can only be fused at very high temperatures, and it cannot therefore be made into castings, but, being soft at a red heat, it can be forged by hammering into any desired shape. It also possesses the valuable property of *welding* when raised to a white heat (about 1500° or 1600° Fah.), by which process two pieces of wrought-iron can be joined together as one, by hammering. Wrought-iron is used for nearly all the moving parts of engines (where it has not been displaced by steel), especially for such parts as are of simple form and which can therefore be easily forged, such as rods, shafts, levers, bolts and nuts, and wherever a strong, tough material is necessary. The tensile strength of ordinary bar iron is about 25 tons per square inch, and of plate iron 22·5 tons per square inch when the stress is in the direction of the fibre, and 20 tons when across the fibre.

STEEL considered chemically is a material between wrought-iron and cast-iron, containing more carbon than wrought-iron and less than cast-iron. It is manufactured either by the addition of carbon to wrought-iron, or by the extraction of carbon from cast-iron. Steel differs from wrought-iron in possessing the property of *tempering*, that is, of hardening by sudden cooling, and again softening by gradual cooling from a high temperature. For some years the distinction between wrought-iron and the softer kinds of steel have become much less marked. The softer kinds of steel contain the least quantities of carbon; they have the same capacity for welding as wrought-iron, but are stronger and tougher. The harder kinds of steel increase in strength, but they are more brittle, less tough, and less easily welded.

An international committee appointed by the American Institute of Mining Engineers in 1876 recommended the following nomenclature:—

1. That all malleable compounds of iron with its ordinary ingredients, which are aggregated from pasty masses, or from piles, or from any forms of iron not in a fluid state, and which will not sensibly harden and temper, and which generally resemble what is called "wrought-iron," shall be called *weld iron.*

2. That such compounds, when they will from any cause harden and temper, and which resemble what is now called "puddled steel," shall be called *weld steel.*

3. That all compounds of iron with its ordinary ingredients, which have been cast from a fluid state into malleable masses, and which will not sensibly harden by being quenched in water while at a red heat, shall be called *ingot iron.*

4. That all such compounds when they will from any cause so harden, shall be called *ingot steel.*

A large firm of makers divides steel into the following four classes:—

1st Class. Extra mild steel. Carbon 0.05 to 0.20 per cent.; tensile strength 25 to 32 tons per square inch; extension 20 to 27 per cent. in 8 ins.; weld, but do not temper; used for boiler plates, ship plates, nails, wire, &c.

2nd Class. Mild steel. Carbon 0.20 to 0.35 per cent.; tensile strength 32 to 38 tons per square inch; extension 15 to 20 per cent.; scarcely weldable, and hardens a little; used for railway axles, tires, rails, guns, and other pieces exposed to heavy strains.

3rd Class. Hard steel. Carbon 0.35 to 0.50 per cent.; tensile strength 38 to 46 tons per square inch; extension 15 to 20 per cent.; do not weld, but may be tempered; used for rails, special tires, springs, guide bars of steam engines, pieces subject to friction, spindles, hammers.

4th Class. Extra hard steel. Carbon 0.50 to 0.65 per cent.; tensile strength 46 to 51 tons per square inch; extension 5 to 10 per cent.; do not weld, but may be strongly tempered; used for delicate springs, files, saws and various cutting tools.

In the above cases the *extension* is taken as a measure of the *toughness* of the material; and it will be noticed that the softer steels are lower in tensile strength, but tougher or more elastic.

CAST STEEL is now being frequently used for machine parts subjected to severe strains, which were formerly made of cast-iron or brass, such as cross-heads, pistons, link motion blocks, eccentric straps, worms, wheels, &c.

MALLEABLE CAST-IRON is made by imbedding the casting in powered red hæmatite; then raising the whole to a bright red heat, and maintaining that heat for from three to five days. depending on the size of the casting. The hæmatite, which is an oxide of iron, parts with some of its oxygen, which unites with and extracts part of the carbon from the cast-iron, and converts it into a tough, malleable material, not unlike a soft steel.

CASE HARDENING WROUGHT-IRON.—The working parts of a machine which are of wrought-iron are frequently case hardened about the joints where the wear is excessive. The process consists in heating the parts to be case hardened in contact with substances rich in carbon, such as bone dust, wood charcoal, leather, prussiate of potash. By this process the wrought-iron combines with a portion of the carbon and becomes converted into steel. The steel thus formed on the surface of the material is hardened by thrusting the part while at a dull red heat into water.

COPPER is a very malleable, ductile material, and may be rolled into plates, and hammered, pressed or bent when cold into any desired shape. It possesses high conductivity or heat-conducting power, and hence is used in the construction of locomotive fire boxes. The ends of copper plates may be joined by brazing so as to be as strong as the original plate. The tenacity of rolled copper is about 15 tons per square inch. It is used largely by marine engineers in the construction of steam and water pipes, but its chief use is in the manufacture of the various alloys which are included under the terms *brass* and *gun metal.*

Owing to the occurrence of one or two serious cases of bursting of copper steam pipes, supposed to have been due to deterioration of the copper through overheating during brazing, copper steam pipes are now made seamless by a process of cold-drawing, or by the electro-deposition of the copper on a rotating iron mandril, accompanied by a patent process of burnishing and compressing the deposited copper by a rotating agate burnisher.

BRASS is the name given to a great variety of the alloys of copper with zinc, tin and lead. Ordinary brass is composed of copper and zinc, in the proportion of about 2 of copper to 1 of zinc. It is yellower in colour, softer, less strong and tough than gun metal.

GUN METAL or BRONZE is much used for cocks, valves, pumps, bearings, &c. Its composition varies for different purposes, thus:

	Copper.	Tin.	Zinc.
Large gun metal castings	87	8	5
Small do. do. (valves, &c.)	90	8	2
Hard gun metal bearings	85	10	5

Chapter IV.

9. PROJECTIONS. Plates I. and II.

The illustrations on these plates are an introduction to the Projection of Solids, the study of which should be pursued further from works on Solid Geometry.

Plan, Section and Elevation.—If I stand in front of a house and sketch it, I make a drawing of the *front elevation* of the house; in other words, my drawing is a representation of it on a vertical plane, or, the picture which I should obtain by looking at the house through a sheet of glass held vertically. If the front wall were entirely removed, exposing the rooms and their contents, a sketch from the front would be a *sectional elevation*. If I stand at the side of the house and sketch it, I get a *side* or *end elevation*. If I were in a balloon right over the house, the sketch of what I could see of it would be a *plan*, or, in other words a representation of it on a horizontal plane. If the whole house were removed and nothing were left but the foundations, a drawing of this would represent the *ground plan*.

A *section* literally means a *cutting*, and sections or cuttings are assumed to be taken through parts of machines to show more clearly on the drawing the construction of the internal parts.

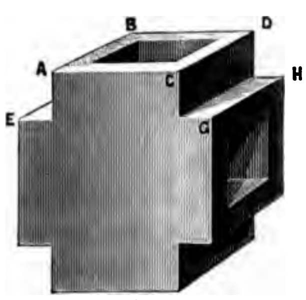


Fig. 12.

The solid shown on Plate I. and illustrated by Fig. 12 is to be drawn to the given dimensions, the dimensions to be taken from a rule marked with inches, eighths and sixteenths; the front elevation first, then the side elevation. The heights of the different lines on the side elevation are obtained by projecting with a T square from the corresponding lines in the front elevation. It will be noticed that one-half of this view is shown in section; that is, one-half of the front of the box is supposed to be cut away to show clearly the thickness of the material, and the shape of the figure internally.

The plan is obtained by projecting corresponding lines from the front elevation. The length *ef* is therefore obtained by projection; the length *ac* or *eg* from the dimensions on the side elevation. The sectional parts in the plate are shown by cross hatching. This is best shown on a student's drawing by a wash of neutral tint.

Plate II. represents a solid, somewhat similar in form to that on Plate I.; the dimensions are, however, a little different, and there are parts projecting from *four* faces of the solid instead of from *two* as on Plate I. The *front elevation* is shown with its base inclined at an angle of 30° with the ground line. This view should be drawn first, and to the dimensions given. All the lines in it may be drawn with the 60° set square. Now draw the side elevation, projecting all the horizontal lines from the front elevation. The vertical lines drawn in position, and to the dimensions given, complete the view. The *plan* is obtained by projecting all vertical lines from the front elevation. The horizontal lines are obtained from the dimensions given on the side elevation. Corresponding points in the various views are lettered similarly.

PROJECTIONS OF A T JOINT. PLATE III.

Fig. 13.

The true shape of the joint will be understood by reference to Fig. 13. First draw the principal centre lines of the figures. Do not crowd the views together nor spread them out too much. Begin with the left hand figure. Draw the top and bottom lines by measuring on each side of the horizontal centre line a distance equal to one-half of $3\frac{3}{4}$ ins. (the extreme length of the figure). Make the lines $2\frac{5}{8}$ ins. long. The parts shown overhanging or projecting from the extremities of the main body of the pipe are called *flanges*. They are used for securing the pipe to another similar flange on another pipe. Holes are drilled in the flanges as shown, through which bolts are passed for bolting the flanges together, and thus making a tight joint.

The figure may be completed from the dimensions given, care being taken to set off the parts at equal distances on each side of the centre line. The remaining views are carefully projected the one from the other; the dimensions required are obtained from the sectional view. The parts shown in section by cross-hatching should be shown by the student by a wash of neutral tint (for cast-iron). The width of *flanges* for steam joints, namely, the distance from the edge of the hole to the edge of the flange, is usually $2\frac{3}{4}$ times the diameter of the bolt. The bolt holes are drilled in the flanges as close to the body of the cylinder as possible, just sufficient room being left for the head of the bolt to clear the side of the cylinder.

Questions.—1. Find the area of the circular opening in the pipe (diameter $1\frac{1}{4}$ ins.)

2. Find the internal diameter for a pipe of double the sectional area of the given pipe $1\frac{1}{4}$ ins. diameter.

NOTE.—The problem (Question 2) is easily solved by a geometrical construction. Draw from any point A, two lines A B and A C $1\frac{1}{4}$ ins. long, at *right angles* to each other; join B C; then B C is the diameter of the circle required, whose area is equal to the *sum* of the areas of the circles whose diameters are A B and A C. Verify your result by arithmetic.

10. VALVES. PLATES IV. AND V.

VALVES are a detail of great importance to the engineer. In some form or other they are continually required to regulate the admission and discharge of steam or water, by opening or shutting the passage through which the fluid flows.

The *seating* of a valve is that part upon which the valve rests or presses when closed. The *face* of a valve is that part of it which comes in contact with the seating. The *valve box, or chamber*, is the part in which the valve works. (See Feed Pump, Plate XXX.) This chamber should always be so constructed as to allow an easy passage of the fluid when the valve is open.

Valves are made of various materials, as wood, leather, indiarubber, iron, gun metal, phosphor bronze.

The *flap valve* is perhaps the simplest form of valve, and may be found in the common well pump. In its rudimentary form it consists merely of a piece of leather secured on one edge, and free to move like a lid turning on a hinge. The leather flap may be stiffened by means of a piece of lead or iron at its back.

A development of the same simple valve, made of brass, and working upon a hinge, and a metal seating, is very commonly used for condensers.

The *conical disc valve*, or puppet valve (Plate IV., and Fig. 14 A and B), is a flat, slightly arched disc of metal, which rests on a circular seating. The face of the valve and the edge of the seating are made conical, to an angle, usually, of 45°, and they are turned and ground to fit one another exactly. The bearing surface of the seating is made narrow—varying from about $\frac{1}{16}$ to $\frac{3}{16}$ in. to ensure tightness of fit. Beneath the disc, in Fig. 1, Plate IV., is seen (best in the plan) three feathers or webs (compare Fig. 14A), which act as a guide to the valve, and enable it to slide up or down vertically, and to return again to its seating when closed.

Fig. 14A.

Sometimes, instead of the feathers, the disc is provided with a spindle for the same purpose, which works through a hole or socket in the seating. This arrangement is shown in Fig. 2 in the Plate, and at Fig. 14B. The knob on the top of the disc is placed there to prevent the valve rising too high from its seating. The knob strikes against a stop, which is always so placed that the valve shall not lift more than $\frac{1}{4}$th of its diameter. This is the amount of lift necessary to make the area of escape round the edge of the valve equal to the area of the hole in the valve seating. A greater lift than this would be useless. With large valves the lift permitted is much less than this, to prevent undue shock of the valve on its seating.

Fig. 14B.

To Draw the Conical Disc Valves.—The student will notice that Figs. 1 and 2, Plate IV., are illustrations of two different valves, as shown also at Fig. 14 A and B.

Fig. 1. The seating for this valve is not shown. Two views are given, the elevation and the plan. The upper part of the elevation, namely, the disc of the valve, may be drawn first, but before this view can be completed the plan must be drawn, showing the feathers in position. Then the lines representing the feathers in the elevation are projected from the plan. The centre lines of the feathers divide the circle into three equal parts; in other words, they make 120° with each other. They may be drawn with the 30° set square.

Fig. 2 shows the valve together with its seating. As before explained, instead of feathers this valve has a spindle which works in a socket provided for it in the seating. The front half of the seating is supposed to be cut away to show the spindle plainly. The plan shows the seating only, with the valve removed. It is shown broken for want of room. The student will draw the circles complete. The valves and seating are made of brass.

Questions.—1. Find the weights of the valves in Figs. 1 and 2.

2. Find the effective area of the opening in the valve seating, Fig. 2.

INDIARUBBER DISC VALVE. PLATE V.

This description of valve is very much used for air pumps and condensers. It consists of an indiarubber disc IRD (Fig. 15), which rests on a grating, and which, owing to its flexibility, readily lifts and permits the water to pass upwards through the grating, and when closed prevents it from returning. A thin cup-shaped guard G is placed above the india-rubber disc, against which it presses during the upward rush of the water through the grating of the seat. The guard itself is provided with holes to lighten it. It is secured to the seating by a brass stud and a hexagon nut. These valves are noiseless, and work well when not too large. It is found to answer better to use several small ones than one large one of this type. Some difficulty has been found with this valve when used in the air pumps of marine engines owing to the fact that the condensed steam contains mineral oil, which acts injuriously upon the indiarubber. Instead of indiarubber, patent valves made of thin rolled sheet phosphor-bronze are now being largely adopted.

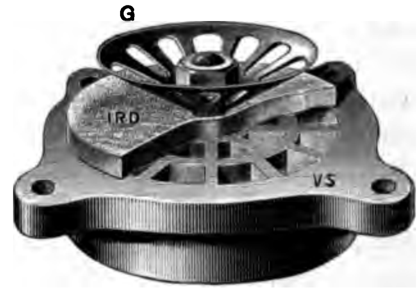


Fig. 15.

To DRAW THE INDIARUBBER DISC VALVE (PLATE V.)—Draw the centre lines, and begin first with the sectional view. Begin at the top and work downwards. It is usually advisable to keep both views going together, as the same dimension, when once obtained, may be used for both cases. The centre lines of the holes in the guard are all drawn with a 60° set-square, and the lines of the grating either with a 60° or 45° set-square.

Questions.—1. Find the solid contents of the indiarubber disc.
2. Find the total area of the waterways in the brass seating.

11. ROD WITH STRAP GIB AND COTTER ENDS. PLATE VI.

When the motion of one part of a machine is required to be transmitted to some other part, the parts are linked together by means of a connecting rod. This might be a simple flat bar with a hole in each end passing over a pin; but such a bar would contain no provision for adjustment when the hole had worn large. In order therefore to provide suitable means of ready adjustment,

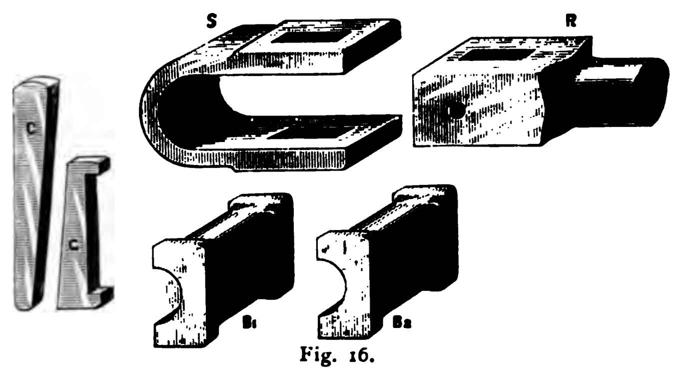


Fig. 16.

various devices are adopted, of which the strap gib and cotter, illustrated in the Plate, is one. Fig. 16 illustrates the appearance of the separate parts. In the Plate the parts are shown in position, "coupled up."

The rod itself (R R), which is of circular section with square butt ends, is made somewhat shorter than the total distance between the pins to be connected, the remainder being made up by the brasses which abut against the flat ends of the rod. The brasses B_1 B_2 are made in two halves, and are kept in position by the U shaped strap S. They have projecting flanges on each side, so as to prevent lateral movement. The strap is secured in position and tightened on the brasses by the gib G and the cotter C, which pass right through the slots in strap and rod as shown. The ends of the gib turn over and grasp the strap to prevent the ends from spreading open. It will be noticed that though the outer edges of the gib and cotter are parallel, their inner edges are slightly tapered, so that when the cotter is struck on the top it acts as a wedge, drawing the strap tightly over the brasses. The cotter bears against the rod, while the gib bears against the two sides of the strap, and it will be seen that the clearance in front of the gib in the rod, and behind the cotter in the strap, is necessary, in order to allow of the movement of the parts which results from tightening up the cotter. To prevent the cotter from shaking loose, it is secured in its place by a small set screw.

The taper of cotters varies from 1 in 24 to 1 in 48 for simple cotters; for cotters with a set screw, from 1 in 8 to 1 in 16.

To Draw the Example.—First lay off the centre lines. The example is to be drawn full size to the dimensions, but the distance between the centres on the drawing need not be 22 ins., because the rod is a plain one, and of the same diameter throughout; hence, to save room it is shown broken off. About 14 ins. between centres (instead of 22 ins.) will do. The rod must be shown broken, and dimensioned 22 ins. It will be noticed that the semi-circular ends of the brasses and straps are not struck from a common centre, owing to the thickness at the ends being greater than that at the sides. The rod and straps are of wrought-iron, and the cotters of steel.

Chapter V.

SCREW THREADS AND BOLTS. Plates VII. and VIII.

(*a*) Let the vertical straight line A A′, Fig. 17, intersect a horizontal plane in A, and let the point P revolve about the fixed point A, in the plane, and at a uniform distance from it; then the path of P is a *circle.*

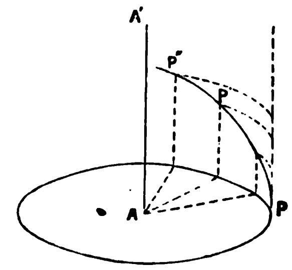


Fig. 17.

(*b*) Let the point P move parallel to A A′; then the path of P is a *straight line.*

(*c*) Let now these two motions be combined; then the point P will describe a curve, P P′ P″, called the *helix* or *screw.*

The distance travelled by the point P in the direction of the axis during one complete revolution is termed the *pitch.*

If a piece of paper be cut as shown in Fig. 18, so that the part marked "circumference" fits exactly round a cylinder, while the part marked "pitch" lies on the cylinder parallel to the axis, then the hypotenuse of the triangle will form a *helix* or *screw* on the surface of the cylinder.

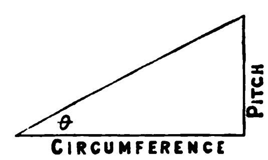


Fig. 18.

The figure of the screw may be described in general terms as consisting of a thread which winds in successive coils round a cylinder. (Illustrated by winding a piece of string upon a pencil or round ruler). The distance between two successive coils, measured parallel with the axis, is called the *pitch.* The pitch is also defined to be the distance travelled by the nut, in the direction of the axis, in making one complete revolution. This distance is also called the *axial pitch.*

The *normal pitch* of a screw is the distance from one thread to the corresponding point on the next coil of the same thread, measured, not parallel to the axis, but in a direction at right angles to the coils.

The counterpart, or complement, of the screwed cylinder is called the *nut* (Fig. 19), and contains a corresponding helical groove, which receives the helical projection of the cylinder. The thread enters the groove by rotation of either element of the pair.

Screws are termed *right-handed* when they require to be turned in a right-handed direction (that is, the direction of rotation of the hands of a watch) in order to make them advance. When they advance by being turned in the opposite direction they are termed *left-handed.* On looking along the axis of a screw, when the visible parts of the thread move onwards toward the right, the screw is right-handed, and *vice versâ.*

When a *single* thread is wound upon the cylinder it is called a *single-threaded* screw. When *two* parallel threads are wound upon it, the screw is termed a *double-threaded* screw, and so on.

In practice, threads are formed by cutting grooves in the bar with a sharp tool, thus leaving the thread standing upon the bar.

Fig. 1, Plate VII., shows a square threaded screw cut out of a 6 ins. bar. Draw a centre line, then lines 3 ins. on each side of the centre for the outside diameter of the thread, then a horizontal line across the bottom of the screw. Below Fig. 1, and on the same centre line, draw a circle 6 ins. diameter, to represent the plan or end of the bar out of which the thread is cut. The pitch of the thread is 1½ inch, therefore mark off along the left hand vertical line distances of 1½ inch. Subdivide this distance into any number of equal parts, say 12, beginning at 0'. Divide the circular plan into the *same* number of equal parts, beginning at 0, and number the divisions in each Fig. as shown. Now, as the thread winds round from 0 to (say) 6 in the plan, it rises from 0' to 6' in the elevation, and by the time it has made one complete circuit, the thread has risen to 12' in the elevation. Draw, therefore, horizontal lines through 0', 1', 2', 3', &c., and raise verticals from 0, 1, 2, 3, &c., to intersect their respective horizontals; thus 1 to 1', 2 to 2', &c. Where these lines intersect we obtain a series of points. Draw through these points a freehand curve, and the required curve of the thread is obtained. Repeat this curve at distances of half the pitch. A ready method of doing this is to trace the curve on a piece of tracing paper, then with a hard pencil or point transfer it to a piece of cardboard or thin wood. Cut it out with a penknife, and finish the edge, if necessary, with sand paper. The remaining curves may be ruled with this templet.

The inner curve, representing the bottom of the thread, is drawn in a similar way; thus, measure inward towards the centre, distances *o a*, *d b* equal to the thickness of the thread 0' 6', and raise verticals. Draw a circular plan with diameter *a b*, and raise verticals from 0, 1, 2,, &c., to intersect horizontals through 0' 1' 2', &c., as before.

It is usual in practice to represent the thread by straight lines, which are a sufficient approximation to the true curve. This is illustrated in Fig. 6, which represents a double-threaded left-handed screw.

To draw the figure, make the outside diameter 3 ins. and the pitch 1 in. The thickness of thread *a c* and depth *a f* are ¼ in. Follow the thread from *a* to *h* or from *g* to *k*. Then *a h* or *g k* equals the pitch.

Fig. 7 shows a common method of representing V threads, the angles being left sharp.

To draw a 3 ins. V-threaded screw, mark off lines *a d*, *l r*, 3 ins. apart. By Table of screw threads there are 3½ threads per inch for a 3 ins. screwed bolt. Divide 1 in. into 3½ parts, or 2 ins. into 7 parts. Take *a b* = one part and mark it along *a d*. Now with a 55° set-square draw *a c*, *c b*, &c., all along *a d*. Make *l o* equal half *a b*. Join *a o*, *b p*, &c., throughout. Then from *o*, *p*, &c., draw V's. Lastly, join *c s*, &c.; *c s* will be not quite parallel to *a o* or *b p*.

Fig. 19.

Figs. 3, 4 and 5 show three views of a 2 ins. hexagonal nut. First draw the outer dotted circle of Fig. 5 to given dimension, and in it describe a hexagon with the 60° set-square. The points *a* and *d* of hexagon, when projected to Fig. 4, give the outer edges of the figure, or the width of nut across angles. Points *b* and *g* give outer edges of Fig. 3, or the width of nut across flats. Make the height of nut same as diameter of bolt, namely, 2 inches.

The cutting off the corners of the nut in a lathe, as shown in the upper part of Fig. 19, is called "chamfering," and gives to the nut a more finished appearance.

With centre *s* (Fig. 3) and radius *r*, cut off the corners at *m″ n″* as shown. The radius *r* depends on the amount required to be taken off the corner; *n″ e* is always small, sometimes nothing at all. In this case all the curves shown on the nuts would touch the top line. When the corner is cut off, a line drawn through *e* will give the highest points of the curves throughout. With the same radius *r*, cut off the corners of Fig. 4. Then a line drawn through *a′* or *d′* gives the lowest points of the curves throughout. The curves are then completed as shown. The diameter of the circle *m n* in Fig. 5 is equal to *m′ n′* Fig. 4, or *m″ n″* Fig. 3, Plate VII.

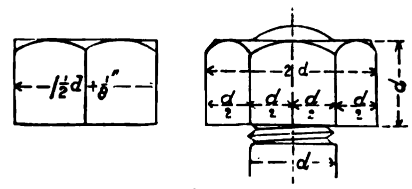
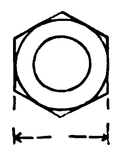
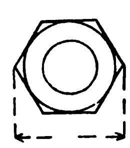


Fig. 20.

The proportions shown in Fig. 20 are approximately correct, and may be used for ordinary drawings.

TABLE IX. PROPORTIONS OF BOLTS AND NUTS.

Diameter of bolt.	Hexagonal Nuts. 		Number of V threads per inch.	Thickness of nut.	Thickness of bolt head.	Area of bolt in sq. inches outside.
½	29/32	1 1/16	12	½	7/16	·1963
⅝	1 3/32	1 5/32	11	⅝	9/16	·3068
¾	1 5/16	1½	10	¾	⅝	·4417
⅞	1½	1 23/32	9	⅞	¾	·6013
1	1 11/16	1 15/16	8	1	⅞	·7854
1⅛	1⅞	2 5/32	7	1⅛	1	·9940
1¼	2 1/16	2⅜	7	1¼	1 3/32	1·227
1⅜	2 3/16	2 9/16	6	1⅜	1 3/16	1·484
1½	2 7/16	2 25/32	6	1½	1 5/16	1·767
1⅝	2 9/16	3	5	1⅝	1 7/16	2·073
1¾	2¾	3 3/16	5	1¾	1 9/16	2·405
1⅞	3	3½	4½	1⅞	1⅝	2·761
2	3 5/32	3⅝	4½	2	1¾	3·141
2¼	3 17/32	4 3/32	4	2¼	2	3·976
2½	3 29/32	4½	4	2½	2 3/16	4·908
2¾	4 3/16	4 13/16	3½	2¾	2 7/16	5·939
3	4 17/32	5 7/32	3½	3	2⅝	7·068
3¼	4⅞	5½	3¼	3¼	2⅞	8·295
3½	5¼	6 1/16	3¼	3½	3 1/16	9·621
3¾	5⅝	6½	3	3¾	3¼	11·04
4	6	6 13/16	3	4	3½	12·56
4¼	6⅜	7⅜	2⅞	4¼	3¾	14·18
4½	6¾	7¾	2⅞	4½	3 15/16	15·9

Number of threads per inch for *square* threads = half the number given for angular threads.

A bolt subjected to a tensile load, that is, a load tending to stretch it, may give way (1) by breaking across at bottom of thread; or (2) by stripping the thread. Usually, a bolt is more likely to break across at bottom of thread than to strip the thread, provided the nut fits the thread properly and that its depth, for V threads, is more than half the diameter of the bolt. The depth of a nut is usually made *equal to* the diameter of the bolt.

The strength of a screwed bolt must be estimated upon its area at its smallest section, namely, at bottom of thread.

$$\text{Dia. of bolt at bottom of thread} = \cdot 9 \text{ outside diameter (approx.)}$$
$$\text{Area of bolt at bottom of thread} = \cdot 8 \text{ area of shank (approx.)}$$

Bolts, under the action of a steady or dead load, may be safely made to carry 4 tons per sq. in.; but when requiring to be well tightened up, or when liable to sudden shocks, as in cylinder cover bolts, a working load of 4,000 lbs. per sq. in. is allowed. For larger bolts, such as marine connecting rod and pedestal bolts, the safe load may be increased to 6,000 lbs. per sq. in.

$$\frac{\text{Load on bolt in lbs.}}{\text{area at bottom of thread}} = \text{load per sq. in.}$$

$$\frac{\text{Total load in lbs.}}{\text{safe load per sq. in.}} = \text{suitable area of bolt at bottom of thread}$$

$$\text{Area at bottom of thread} \div 0{\cdot}8 = \text{outside area of bolt}$$

$$\text{Area at bottom of thread} \times \text{safe load per sq. in.} = \text{safe working load}$$

A suitable outside diameter d for shank or body of screwed bolt (see Fig. 1, Plate VIII.) may be obtained from the following formulæ :—

$$\text{(1.)} \quad d \quad \text{(for small bolts)} = \frac{\sqrt{\text{total load in lbs.}}}{50}$$

$$\text{(2.)} \quad d \quad \text{(for large bolts)} = \frac{\sqrt{\text{total load in lbs.}}}{60}$$

Exercise 1.—Draw a bolt and nut, as in Fig. 1, Plate VIII., full size, to carry with safety a load of 3,600 lbs.; bolt to be well tightened up. Use formula (1).

Exercise 2.—Draw, full size, a connecting rod bolt and nut, as shown in Fig. 2, Plate VIII., to sustain a maximum stress of 24 tons. Use formula (2).

13. RIVETS AND RIVETED JOINTS. Plate IX.

Boiler plates are joined together at their edges by riveting. Bolts and nuts are used as fasteners in cases where the straining force is a tensile one, tending to stretch the bolt and to strip off the head and nut. Rivets are used when the force is a shearing one, tending to cut through the body of the fastener at right angles to its axis, as is the case with boiler plates. The plates to be connected have holes punched or drilled in them, and they are then made to overlap one another so that the holes coincide. The rivet, which has been made red-hot at the end, is then inserted in the hole through the two plates, and a second head is formed by hammering or by pressure of a machine. As the rivet cools, it contracts and grips the two plates very securely.

The various forms of rivets and rivet heads are shown in the Plate (IX.), with their proportions. The figures given indicate the proportion of the parts to the diameter d of the rivet. The first example shows the rivet in position, with its proportions before the second head has been formed. In the second example it will be noticed that the hole in the plate tapers inwards towards the middle of the rivet. This taper is due to punching, and the plates are so placed as to bring the smaller ends of the holes together. When the hole is drilled, its sides are parallel.

Prof. Unwin gives the following rule for the relation between diameter of rivet d, and thickness of plate t, viz., $d = 1{\cdot}2\sqrt{t}$.

Riveted joints are either *lap* joints or *butt* joints. In the *lap joints* the plates overlap, and are secured by one, two, or three rows of rivets, and called respectively single-, double-, or triple-riveted joints. In the *butt joints* the plates are placed edge to edge. The two abutting edges are then covered by a strip of plate called a strap or cover plate, either on one side or both sides, and each plate is then securely riveted to the straps by one, two, or three rows of rivets as required.

THE COMBINED LAP AND BUTT JOINT is one which has been adopted in locomotive boiler construction. It consists of a single-riveted lap joint with a cover plate, the whole being secured by three rows of rivets, as shown, Plate IX.

THE ANGLE IRON JOINT illustrates a method of joining two plates at right angles with one another, such as the two sides of tank. The angle iron itself is a solid strip rolled to the shape of a right angle. Its thickness is about the same as that of the plates to be connected, and its flanges taper slightly towards the root. The plates are secured to the flanges by riveting.

Instead of the angle iron, the plates may be connected by "flanging," or turning over the edge of one of the plates, and riveting, as in Fig. 21.

NOTE.—The rivet is here subjected to a shearing stress. If the other plate had been flanged instead, the rivet would be in tension.

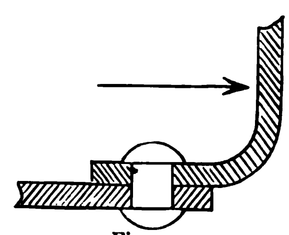


Fig. 21.

THE T IRON JOINT answers the double purpose of a cover plate and a stiffener.

The following tables, from Seaton's "Manual of Marine Engineering," give the pitch of rivets as found in general practice for wrought-iron plates and rivets :—

TABLE X. LAP JOINTS, SINGLE-RIVETED.

Thickness of plate.	Diameter of rivet.	Pitch of rivets.	Breadth of lap.
$\frac{1}{4}$ inch.	$\frac{1}{2}$ inch.	$1\frac{1}{8}$ inch.	$1\frac{1}{2}$ inch.
$\frac{5}{16}$,,	$\frac{5}{8}$,,	$1\frac{1}{2}$,,	$1\frac{7}{8}$,,
$\frac{3}{8}$,,	$\frac{3}{4}$,,	$1\frac{3}{4}$,,	$2\frac{1}{4}$,,
$\frac{7}{16}$,,	$\frac{3}{4}$,,	$1\frac{3}{4}$,,	$2\frac{1}{4}$,,
$\frac{1}{2}$,,	$\frac{7}{8}$,,	$2\frac{1}{8}$,,	$2\frac{5}{8}$,,
$\frac{9}{16}$,,	$\frac{15}{16}$,,	$2\frac{1}{8}$,,	$2\frac{13}{16}$,,
$\frac{5}{8}$,,	1 ,,	$2\frac{1}{4}$,,	3 ,,
$\frac{11}{16}$,,	$1\frac{1}{8}$,,	$2\frac{1}{2}$,,	$3\frac{1}{4}$,,
$\frac{3}{4}$,,	$1\frac{1}{8}$,,	$2\frac{1}{2}$,,	$3\frac{1}{4}$,,

TABLE XI. LAP JOINTS, DOUBLE-RIVETED (ZIGZAG).

Thickness of plate.	Diameter of rivet.	Pitch of rivets.	Breadth of lap.
$\frac{1}{2}$ inch.	$\frac{3}{4}$ inch.	$2\frac{1}{2}$ inches.	$3\frac{3}{4}$ inches.
$\frac{9}{16}$,,	$\frac{7}{8}$,,	3 ,,	$4\frac{1}{4}$,,
$\frac{5}{8}$,,	$\frac{7}{8}$,,	3 ,,	$4\frac{1}{4}$,,
$\frac{11}{16}$,,	1 ,,	$3\frac{3}{8}$,,	$4\frac{3}{4}$,,
$\frac{3}{4}$,,	$1\frac{1}{16}$,,	$3\frac{1}{2}$,,	$4\frac{3}{4}$,,
$\frac{13}{16}$,,	$1\frac{1}{8}$,,	$3\frac{3}{4}$,,	$5\frac{1}{4}$,,
$\frac{7}{8}$,,	$1\frac{1}{8}$,,	$3\frac{1}{2}$,,	$5\frac{1}{4}$,,
$\frac{15}{16}$,,	$1\frac{1}{4}$,,	4 ,,	$5\frac{1}{2}$,,
1 ,,	$1\frac{1}{4}$,,	4 ,,	$5\frac{1}{2}$,,
$1\frac{1}{16}$,,	$1\frac{1}{4}$,,	4 ,,	$5\frac{1}{2}$,,

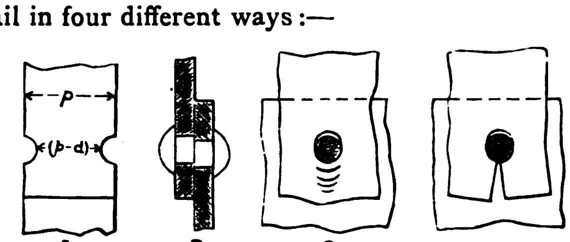
TABLE XII. BUTT JOINTS, DOUBLE STRAPS, DOUBLE-RIVETED (ZIGZAG).

Thickness of plate.	Diameter of rivets.	Pitch of rivets.	Breadth of straps.	Thickness of straps.
$\frac{3}{4}$ inch.	$\frac{7}{8}$ inch.	$3\frac{1}{2}$ inches.	$8\frac{3}{4}$ inches.	$\frac{1}{2}$ inch.
$\frac{13}{16}$,,	$\frac{15}{16}$,,	$3\frac{3}{4}$,,	$9\frac{3}{8}$,,	$\frac{1}{2}$,,
$\frac{7}{8}$,,	1 ,,	4 ,,	10 ,,	$\frac{9}{16}$,,
$\frac{15}{16}$,,	$1\frac{1}{16}$,,	$4\frac{1}{4}$,,	$10\frac{5}{8}$,,	$\frac{5}{8}$,,
1 ,,	$1\frac{1}{8}$,,	$4\frac{1}{2}$,,	$11\frac{1}{4}$,,	$\frac{5}{8}$,,
$1\frac{1}{16}$,,	$1\frac{3}{16}$,,	$4\frac{3}{4}$,,	$11\frac{7}{8}$,,	$\frac{11}{16}$,,
$1\frac{1}{8}$,,	$1\frac{1}{4}$,,	5 ,,	$12\frac{1}{2}$,,	$\frac{3}{4}$,,
$1\frac{3}{16}$,,	$1\frac{5}{16}$,,	$5\frac{1}{4}$,,	$13\frac{1}{8}$,,	$\frac{3}{4}$,,
$1\frac{1}{4}$,,	$1\frac{5}{16}$,,	$5\frac{1}{16}$,,	13 ,,	$\frac{13}{16}$,,

PROPORTIONS OF RIVETED JOINTS.

Riveted joints are the weakest parts of a new boiler, owing to the fact that the plates have been perforated with holes for the insertion of the rivets, the section of the solid plate being thereby reduced.

A riveted joint may fail in four different ways:—



A B C D

Fig. 22.

(i.) By tearing the plate between the rivet holes, as in Fig. A.
(ii.) By shearing the rivet, as in Fig. B.
(iii.) By crippling the plate between its edge and the rivet holes, as in Fig. C.
(iv.) By breaking the plate between the edge and the rivet holes, as in Fig. D.

Let t = thickness of plate; d = diameter of rivet;

a = area of rivet = $d^2 \times \cdot 7854$;

p = pitch of rivets; F_t = tensile strength of plate per sq. in.; F_s = shearing strength of rivets per sq. in.

1. SINGLE-RIVETED LAP JOINT.—Consider a strip of plate whose breadth is equal to the pitch, or distance from centre to centre of rivets (Fig. 22 A), and let F_t = tensile strength, and F_s = shearing strength of wrought-iron per sq. in.

Then, resistance of plate to tearing between rivet holes = area of section × strength per sq. in. = $(p - d)\, t \times F_t$.

Resistance of (two halves, or one whole) rivet to shearing = area of section × strength per sq. in. = $a \times F_s$.

But in order that the plates and rivets may be of equal strength,

$$(p - d)\, t \times F_t = a \times F_s.$$

For wrought-iron F_s and F_t are about equal.

$$\therefore (p - d)\, t = a$$

$$\text{or, } p = \frac{a + d\,t}{t} = \frac{a}{t} + d;$$

or, in words,

$$\text{pitch of rivets} = \frac{\text{area of rivet}}{\text{thickness of plate}} + \text{diameter of rivet.}$$

Thus, find the pitch of rivets for $\frac{1}{2}$ in. plate, $\frac{7}{8}$ rivets, so that strength of plates and rivets may be equal.

$$\text{Then, } p = \frac{a}{t} + d$$

$$= \frac{\cdot 875 \times \cdot 875 \times \cdot 78}{\cdot 5} + \cdot 875$$

$$= 2\tfrac{1}{8} \text{ ins.}$$

For steel rivets and plates the value of p would be—

$$p = \frac{F_s}{F_t} \cdot \frac{a}{t} + d$$

$$= \frac{23}{28} \cdot \frac{a}{t} + d$$ ... See Board of Trade Rules, page 44

where 23 tons = shearing strength of steel, and 28 tons = tensile strength.

2. DOUBLE-RIVETED LAP JOINT.—Taking breadth of strip = p as before, there are two rivets to resist shearing;

$$\text{or, } (p - d)\, t = 2\,a$$

$$p = \frac{2\,a}{t} + d$$

or, for steel rivets and plates,

$$p = \frac{23}{28} \cdot \frac{2\,a}{t} + d$$

3. For Butt Joints, with double cover plates and *single*-riveted, it will be seen that each rivet requires to be sheared through *two* sections instead of through one as in the lap joint.

$$\text{Hence} \quad (p - d)\,t = 2\,a$$

$$\text{or,} \quad p = \frac{2\,a}{t} + d$$

which is the same as for double-riveted lap joints.

4. For Butt Joints, with double cover plates and *double*-riveted, we have two rows of rivets for each plate, and each rivet is in double shear; hence the equation becomes—

$$(p - d)\,t = 4\,a$$

$$\text{or,} \quad p = \frac{4\,a}{t} + d$$

When rivets are in double shear, the shearing area is taken as 1·75 times that in single shear, instead of 2 times, which is their theoretical value.

The strength of a plate to resist failure by crushing in front of the rivet, Fig. 22 C, depends upon the thickness of the plate t, and the diameter d of the rivet. The *bearing area* of the rivet is estimated normally to the pressure, and is equal to $d\,t$. The value of the crushing strength of the plate is not known.

The failure of the plate by breaking in front of the rivet, as in Fig. 22 D., depends upon the distance between the edge of the hole and the edge of the plate. Experiment has shown that when this distance is made equal to the diameter of the rivet as a minimum, the plate is sufficiently strong.

Ex. 1. Compare the tensile strength of the plate and the shearing strength of the rivets in a single-riveted lap joint $\frac{3}{8}$ in. plate, $\frac{3}{4}$ in. rivet, pitch $1\frac{3}{4}$ in.

Assuming resistance of plate to tensile stress and of rivet to shearing stress to be equal, then the strength of plate and rivet in above joint are in proportion to their area.

$$\begin{aligned} \text{Area of plate between holes} &= (p - d)\,t \\ &= (1\tfrac{3}{4} - \tfrac{3}{4})\,\tfrac{3}{8} \\ &= \tfrac{3}{8}, \text{ or } \cdot375 \text{ sq. ins.} \\ \text{Rivet area, in width of plate equal to the pitch} &= \text{two halves, or one whole rivet} \\ &= \cdot75 \times \cdot75 \times \cdot7854 \\ &= \cdot4417 \text{ sq. ins.} \end{aligned}$$

Here, area of plate : area of rivet, as 37·5 : 44; or area of plate $= \dfrac{37\cdot5 \times 100}{44} = 85$ per cent. of area of rivet.

Ex. 2. With same dimensions as Example 1, compare the plate and rivets when both are steel.

Taking tensile strength of steel = 28 tons per sq. in., and shearing strength = 23 tons,

Then, resistance of plate to tearing between rivet holes—

$$\begin{aligned} (p - d)\,t \times 28 &= (1\tfrac{3}{4} - \tfrac{3}{4})\,\tfrac{3}{8} \times 28 \\ &= 10\cdot5 \text{ tons} \end{aligned}$$

Resistance of rivets to shearing—

$$\begin{aligned} \text{Area of rivet} \times 23 &= \cdot4417 \times 23 \\ &= 10\cdot159 \text{ tons} \end{aligned}$$

Then, strength of plate is to strength of rivet as 10·5 : 10·159; or, rivets and plate are about equal.

14. BOILER STAYS. PLATE X.

Under this head we will consider the nature of the stresses which occur in steam boilers, and refer to some of the problems which arise in this connection in ordinary practice.

In the early days of steam boiler construction, the pressures used were 3 or 4 lbs. to the square inch only, and boilers were constructed without regard to suitability of form to support internal pressure. But as steam pressures began to increase, increased attention to this point became necessary. To-day 150 lbs. pressure to the sq. inch is not uncommon, and to carry this safely the strongest possible form of boiler must be adopted.

The sphere is the strongest form of vessel to resist internal pressure, but there are many practical reasons which prevent its being used for the purpose. Next to the sphere the cylindrical form of boiler is the simplest and strongest, and it is now almost universally adopted.

RESISTANCE OF CYLINDRICAL VESSELS WHEN SUBJECTED TO INTERNAL PRESSURE.—Let Fig. 23 represent a thin cylindrical vessel subjected to internal pressure.

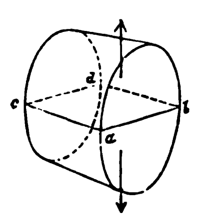


Fig. 23.

Let p = internal pressure per square inch, d = diameter of cylinder, t = thickness of plate, and l = length of cylinder.

It is evident that the internal pressure p is acting radially from the centre on every part of the internal circumference of the shell, but the resultant force tending to separate the cylinder into two parts through the plane $a\ b\ c\ d$, can be shown to be equal to $p \times d \times l$.

The area of the material to resist this tendency to burst along the lines $a\ c$ and $b\ d$ = $(a\ c + b\ d)\ t$; $= 2\ t\ l$. Hence the stress per sq. in. on the plate

$$= \frac{\text{load}}{\text{area}} = \frac{p\ d\ l}{2\ t\ l} = \frac{p\ d}{2\ t} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (1)$$

From which we learn that the stress on the material increases as the pressure and the diameter increase, and that if we either double the pressure or double the diameter, we also double the stress on the plate.

In practice the section would be taken through the weakest part, which, in a new boiler, is through the rivet holes. The strength of a double riveted joint is usually taken at 70 per cent. of the solid plate, and of a single riveted joint 56 per cent.

Again, suppose we require to find the pressure tending to tear the boiler in two in a plane perpendicular to the axis; in other words, tending to blow the end off.

$$\text{Area of end} = d^2 \times \cdot 7854.$$

$$\text{Total pressure on end} = (d^2 \times \cdot 7854)\ p.$$

The area of the material to resist this tendency (neglecting deductions for rivet holes) = circumference of shell $\times$ thickness of plate, $= d \times 3 \cdot 1416 \times t$. Hence the stress per sq. in. on the plate

$$= \frac{\text{load}}{\text{area}} = \frac{(d^2 \times \cdot 7854)\ p}{d \times 3 \cdot 1416 \times t} = \frac{p\ d}{4\ t} \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

Comparing this result with that obtained in (1) we see that a cylindrical boiler is twice as strong transversely as longitudinally. For this reason the longitudinal joints are made stronger than the transverse. Thus the longitudinal joints of boilers are double-riveted, while the transverse or circumferential joints only require a single row of rivets; or, in high pressure boilers, the longitudinal joints are treble-riveted, while the transverse joints are double-riveted.

Ex. 1. Find the stress per sq. in. longitudinally on the plate of a cylindrical vessel 5 ft. in diameter, working at 100 lbs. pressure; thickness of plate $\frac{1}{2}$ in. iron.

$$\text{By (1) stress} = s = \frac{p\,d}{2\,t} = \frac{100 \times 5 \times 12}{2 \times \frac{1}{2}} = 6000 \text{ lbs. per sq. in.}$$

Similarly we may obtain any one of the other terms, pressure of steam p, diameter of boiler d, or thickness of plate t, having given the value of the three remaining terms;

$$\text{For } p = \frac{2\,t\,s}{d};\ d = \frac{2\,t\,s}{p};\ t = \frac{p\,d}{2\,s}$$

The factor of safety necessarily varies according to the quality of material, workmanship, and design of a boiler. The Board of Trade allow a factor of safety of 5 for well constructed new boilers. In ordinary practice it is more usual to take 6 as the factor.

$$\text{Then, } \frac{\text{bursting pressure}}{6} = \text{working pressure.}$$

Let S = the ultimate tensile strength of the material in lbs. per sq. in.
t = thickness of the plate in inches.
d = diameter of boiler in inches.
P = the bursting pressure of the steam in lbs. per sq. in.

Then, as before (equation 1),

$$S = \frac{P\,d}{2\,t};\ \text{or } P\,d = 2\,t\,S$$

The value of S allowed by the Board of Trade is—

47,000 lbs. tensile strength of iron with the grain;
40,000 lbs. tensile strength of iron across the grain;
47,000 lbs. shearing strength of iron rivets;
28 tons tensile strength of steel plates;
23 tons shearing strength of steel rivets.

Ex. 2. Find the safe working pressure for a cylindrical boiler 9 ft. diameter made of $\frac{5}{8}$ in. iron plate, with double-riveted lap joint seams; allowing a factor of safety of 6.

$$\text{By equation } P\,d = 2\,t\,S$$

$$\therefore \text{ bursting pressure } P = \frac{2\,t\,S}{d}$$

$$\text{or, } P = \frac{2 \times \frac{5}{8} \times 47000}{108}$$

$$= 544 \text{ lbs.}$$

This is the bursting pressure supposing there were no seams. Allowing 70 per cent. of the strength of solid plate for double-riveted joints, also taking factor of safety 6, we have :—

$$\text{Safe working pressure} = \frac{544}{6} \times \frac{70}{100} = 63{\cdot}47 \text{ lbs. per sq. in.}$$

Before subjecting boilers to high steam pressures, they are first tested by hydraulic pressure to *twice* the pressure at which they are required to work.

The strength of the joint will depend, of course, upon a proper proportion being observed between the thickness of the plate and the diameter and pitch of the rivets, whether the holes in the plate have been drilled or punched, and whether the workmanship is good or bad.

The percentage of the strength of *plate* at the joint as compared with the solid plate

$$= \frac{(\text{pitch} - \text{diameter of rivet}) \times 100}{\text{pitch}}$$

The percentage of the strength of *rivets* as compared with the solid plate

$$= \frac{(\text{area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{pitch} \times \text{thickness of plate}}$$

Then the smaller of the two percentages is the relative strength of the joint.

Ex. 1. Find the percentage of the strength of *plate* between the rivet holes for a plate with $\frac{7}{8}$ rivets, 3 ins. pitch.

By equation, $\frac{(\text{pitch} - \text{dia. of rivet}) \times 100}{\text{pitch}} = \frac{(3 - \frac{7}{8}) \times 100}{3} = 70{\cdot}8$ per cent.

Ex. 2. Find the percentage of strength of *rivets* as compared with the solid plate in a double-riveted lap joint plate, $\frac{7}{8}$ rivets, 2 rows, 3 ins. pitch.

$$\frac{(\text{area of rivet} \times \text{No. of rows}) \times 100}{\text{pitch} \times \text{thickness of plate}} = \frac{(\cdot 6 \times 2) \times 100}{3 \times \cdot 625} = 64 \text{ per cent.}$$

When the rivets are subjected to *double shear*, as in butt joints with double straps, multiply the above result for strength of rivet by 1·75.

Strength of Furnaces of Circular Section to Resist Collapse.—Cylindrical vessels subjected to *external* pressure should be as perfectly circular in section as possible, because such external pressure tends to increase the slightest variation from the circle, and to cause the failure of the tube by distortion rather than by crushing of the material. Hence, welded or butt joints, which do not interfere with the circular form of the section, are superior to lap joints for boiler furnace tubes.

The following empirical formula gives the proportions recommended by the Board of Trade for boiler furnaces :—

$$\text{Safe working pressure} = \frac{90{,}000 \times t^2}{(L + 1) \times d}$$

Lloyd's formula is,

$$\text{Safe working pressure} = \frac{89{,}600 \times t^2}{L \times D}$$

When D = external diameter of furnace in ins.; L = length in feet (if rings are fitted, length between rings to be taken) : t = thickness of plate in ins.

Each of the above authorities specify that in no case shall the working pressure obtained by the above formula exceed $\frac{8000 \times t}{D}$ lbs. per sq. in.

The thickness of furnace plates should not exceed $\frac{1}{2}$ inch, and the strength of the tube can be kept within safe limits by inserting a stiffening ring, which acts also as an expansion joint, and which also virtually shortens the length of the tube. For this purpose are used the Bowling expansion hoops, Fig. 24; Adamson's flanged seam, Fig. 25; and corrugated flues. (See Plate XXXVIII.)

The Bowling expansion hoop is weldless, and made in iron or steel, varying in thickness from $\frac{3}{8}$ to $\frac{9}{16}$ in. iron, and from $\frac{6}{20}$ to $\frac{9}{20}$ in. steel. The advantage of this joint is that the rivets and double thicknesses of plate are removed from the action of the fire.

The Adamson flanged seam consists of a solid forged ring, $\frac{3}{8}$ to $\frac{1}{2}$ in. thick, placed between, and riveted to, the flanged ends of the tubes.

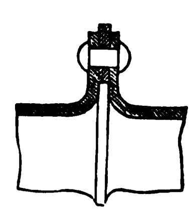


Fig. 24.
Bowling Expansion Hoop.

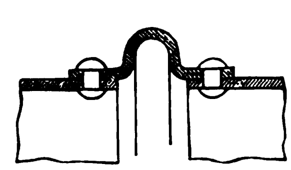


Fig. 25.
Adamson's Flanged Seam.

CORRUGATED FURNACES (shown in the Example, Plate XXXVIII.) are designed from the following formula :—

$$\frac{1000 \times (t - 2)}{D} = \text{working pressure in lbs. per sq. in.}$$

where t = thickness of plate in sixteenths of an inch, D = greatest diameter of furnace in ins. The corrugations are $1\frac{1}{2}$ ins. deep, and 6 ins. pitch.

Corrugated furnaces are now being very largely used. They are much stronger and more flexible, provide a larger heating surface, and are capable of being made thinner than the ordinary furnace.

FLAT SURFACES.—Flat surfaces when required to resist pressure must be well stayed. Fig. 26 represents part of a stayed flat surface. The extent of the load carried by each stay is equal to the pressure on a square whose area $= d \times d$, and the sectional area of the stay must be made sufficient to carry this load. If p = pressure in lbs. per sq. in.; a = area of stay; f = 5000 lbs. = working tensile strength of stay; then load on each stay $= p\,d^2$, and strength of stay $= a\,f$.

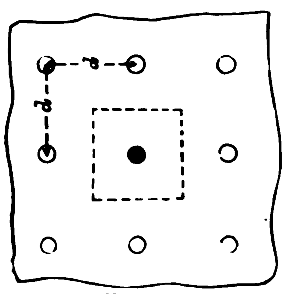


Fig. 26.

$$\text{Therefore } a\,f = p\,d^2; \text{ or, } a = \frac{p\,d^2}{f}$$

Ex. A flat surface is supported by stays $\frac{7}{8}$ in. least diameter, 4 ins. pitch; what pressure of steam may the boiler work at, if safe load on stay be taken = 5000 lbs. per sq. in.?

From above equation,

$$p = \frac{a\,f}{d^2} = \frac{\cdot875 \times \cdot875 \times \cdot7854 \times 5000}{4 \times 4} = \frac{3000}{16} = 187\frac{1}{2} \text{ lbs.}$$

The Board of Trade give the following formula for the strength of flat plates supported by stays :—

$$p = \frac{C \times (T + 1)^2}{S - 6}$$

Where T = thickness of plate in sixteenths,
S = area of surface supported by one stay in sq. ins.,
p = safe working pressure in lbs. per sq. in.

The constant C has the following value for iron plates :—

	Plates not exposed to fire.	Plates with steam on one side and fire on the other.	Plates with water on one side and fire on the other.
Stays, with nuts and washers at least three times diameter of stay and $\frac{2}{3}$ thickness of plates	100	60	...
Stays with nuts only	90	54	...
Stays screwed into plate and nutted ...	...	...	80
Stays screwed into plate and riveted	...	36	60

For *steel* plates, twenty-five per cent. to be added to the above constants, where nuts are fitted to the stays, and ten per cent. where no nuts are fitted.

Fig. 27 illustrates a copper stay used to stay the firebox of a locomotive. These stays are screwed into the walls of the shell and firebox, and riveted over at each end. They are commonly $\frac{7}{8}$ in. diameter in the smallest section, with a pitch of from 4 to $4\frac{3}{4}$ ins. When stays are spaced much more widely than this, the plates are very liable to bulge between the stays to such a degree as to widen the hole and disengage the screw threads.

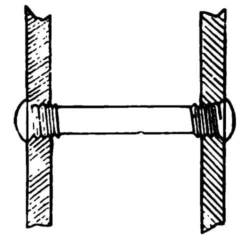


Fig. 27.

DOG OR GIRDER STAY, Plate X., Fig. 28.—The flat crowns of the firebox in locomotive and portable boilers, and the tops of combustion chambers in marine boilers, are supported by flat wrought-iron bars or girder stays, spanning the chamber and resting firmly on the front and back plates, and supporting the crown plate by a series of stay bolts.

Fig. 28.

In locomotives the stays are forged solid, and have 10 or 12 stay bolts. In the illustration given, the stay supports the flat top of the combustion chamber of the marine boiler, Plate XXXVIII. It is made of two plates, side by side, separated by washers and riveted together. A clear space of $1\frac{1}{2}$ ins. is left between the stay and the plate, to allow of a free circulation of the water, and for cleaning out.

Instead of girder stays, the flat top of the firebox or combustion chamber is sometimes attached to the shell by wrought-iron or steel bars. The stays, Fig. 29, are forged with an eye in each end, and are secured to the shell between double angle-iron by means of a pin, and to the combustion

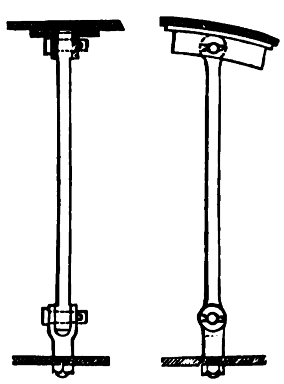


Fig. 29.

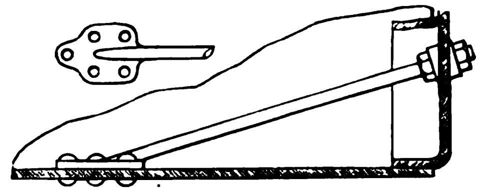


Fig. 30.

chamber by a pin passing through a forked or double-eyed piece, which is screwed into the plate and secured by a nut. A *palm stay*, Fig. 30, is also frequently used to stay curved surfaces.

GUSSET STAYS, Fig. 31 and Plate X., are used to support the flat end plates of cylindrical boilers to prevent them from bulging by the internal pressure. The gusset plate, which passes

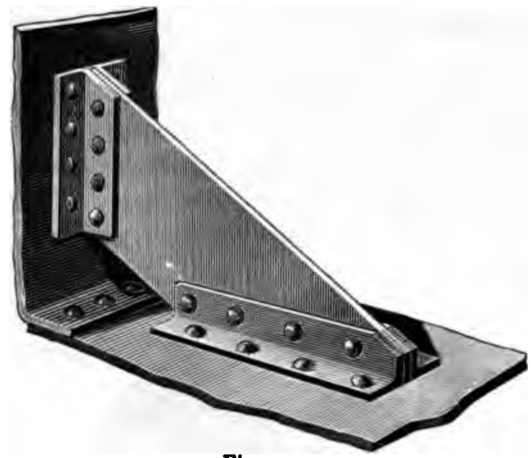


Fig. 31.

diagonally from the end plate to the cylindrical shell, is securely riveted at each end between pieces of angle-iron, which are themselves riveted to the plates.

15. JOINTS AND CONNECTIONS. PLATE XI.

The COTTERED JOINT is an arrangement used for connecting air pump rods, &c., by means of a flat tapered piece called a cotter, which passes through grooves in both rods. The usual proportions allowed for such joints are given on the drawing, and it will be observed that all the dimensions are proportional to d, which is the diameter of the rod in the socket. Where dimensions are not given the student will use his own judgment. The cotter is tapered on one side only, and $\frac{1}{8}$ in. space, or draft, is allowed in the grooves to permit of tightening up. A split pin is inserted to prevent the cotter from slipping back. The cotter is steel, the other parts wrought-iron.

RESISTANCE OF COTTERED JOINTS.—The rods when so placed as to be alternately in compression and tension, would be most likely to break in one of two ways, assuming that the rod itself is stronger than the joint for the sake of stiffness :—

1. Through the weakest section of the rod A (see Plate XI.), namely, where its area is reduced by the cotter hole.

2. By shearing through the cotter at $a\ a$, $c\ c$.

Considering transverse section of rod at weakest part. If d = diameter of rod A in the socket, and t = thickness of cotter, then

$$\text{effective area } a \text{ at section} = \text{area of rod} - \text{area of cotter hole}$$

$$a = {\cdot}78\ d^2 - d\ t$$

To find d for a given load P on bar :—Let safe stress per sq. in. for rod = 6000 for iron, 8000 for steel ; then

$$\text{effective area} = \frac{\text{load}}{\text{safe stress per sq. in.}}$$

$${\cdot}78\ d^2 - d\ t = \frac{P}{6000}$$

If $t = \cdot 25\, d$ for steel, and $\cdot 3\, d$ for iron,

then for iron rod and steel cotter

$$\cdot 78\, d^2 - \cdot 25\, d^2 = \frac{P}{6000}$$

$$\cdot 53\, d^2 = \frac{P}{6000}$$

$$d = \sqrt{\frac{P}{6000 \times \cdot 53}}$$

$$= \frac{\sqrt{P}}{56}$$

that is, diameter of rod at weakest section = the square root of the load $\div$ 56.

Ex. Find the diameter at the weakest section of a cottered bar to carry safely a load of 5000 lbs.

$$\text{Diameter} = \frac{\sqrt{\text{load}}}{56} = \frac{\sqrt{5000}}{56} = 1\tfrac{1}{4} \text{ ins.}$$

Considering the shearing area of the cotter at *a a*, *c c*.

Let b = breadth of cotter, and t = thickness; then area of sections to resist shearing = $2\, bt$. (The value of this shearing area, according to Rankine, $= 2\, bt \times \frac{2}{3}$ owing to accompanying bending action with loose cotter.)

Also let shearing strength of steel = $\frac{5}{4}$ tensile strength of iron;
then, for cotter and rod to be of equal strength,

$$(2\, bt \times \tfrac{2}{3})\, 5 = (\cdot 78\, d^2 - dt)\, 4.$$

If t be taken $= \cdot 25\, d$; then

$$2\, b \times \cdot 25\, d \times \tfrac{10}{8} = (\cdot 78\, d^2 - \cdot 25\, d^2) \times 4$$

$$\cdot 5\, bd = \cdot 53\, d^2 \times 1\cdot 2$$

$$b = 1\cdot 27\, d = 1\tfrac{1}{4}\, d \text{ nearly,}$$

that is, breadth of cotter = $1\frac{1}{4}$ diameter of rod. But diameter of rod in above example = $1\frac{1}{4}$ ins.; hence dimensions of suitable cotter = $1\frac{1}{4} \times 1\frac{1}{4} = 1\frac{9}{16}$ ins. breadth, and $\cdot 25\, d = \frac{1}{4} \times 1\frac{1}{4} = \frac{5}{16}$ in. thickness.

KNUCKLE, OR FORKED JOINT.—Plate XI. The drawing shows three views of a knuckle joint, with the ordinary proportions marked upon them. All the parts are dimensioned proportionally to *d*, the diameter of the rod. This is a very common and useful form of joint, and it permits of a motion of the rods relatively to each other in a plane at right angles to the axes of the pin. The pin is rigidly fixed in the forked end of B by means of a smaller slightly tapered pin *p*, driven into a hole which is drilled, half in the fork, and half in the large pin, as shown. The octagonal parts on the rods are mere ornament, and are done in the forging. The lines representing the corners of the octagon are obtained by projection from the end view.

The relative strengths of the parts of this joint may be considered as follows:—

(i.) To find the diameter of the rods A or B to carry a given load:—If P = load; a = area of rod; f = working tensile stress per sq. in. = 8000 when rod is subjected to tension only, but f = 6000 when the rod is subjected to both tension and compression;

$$a = \frac{P}{f} = \frac{\text{load}}{6000}$$

From which the diameter may be obtained thus:—

$$\cdot 7854\, d^2 = \frac{\text{load}}{6000}$$

$$d^2 = \frac{\text{load}}{6000 \times \cdot 7854}$$

$$d = \frac{\sqrt{\text{load}}}{70} \text{ nearly.}$$

(ii.) To find diameter of pin:—Shearing takes place through two sections of the pin, namely at c and c^1 (Plate XI.); therefore, if d_1 = diameter of pin, and f_1 = shearing strength, then, resistance of pin to shearing $= \cdot 78\, d_1^2 \times 2 \times f_1$. As the pin is more or less loose in the joint, there is an accompanying bending action, which, according to Rankine, makes resistance of section for circular sections $= \cdot 78\, d_1^2 \times 2 \times f_1 \times \frac{3}{4}$. But in order that shearing strength of pin may equal tensile strength of rod, we have

$$\cdot 78\, d_1^2 \times 2 \times f_1 \times \tfrac{3}{4} = \cdot 78\, d^2 \times f;$$

where d = diameter of rod, and f = tensile strength.

Then, if $\dfrac{\text{tensile strength}}{\text{shearing strength}} = \dfrac{f}{f_1} = \dfrac{5}{4}$; the equation becomes

$$\cdot 78\, d_1^2 \times 2 \times \tfrac{3}{4} \times 4 = \cdot 78\, d^2 \times 5$$

$$d_1^2 = d^2 \times \tfrac{5}{6}$$

$$d_1 = \cdot 9\, d,$$

that is, diameter of pin $= \cdot 9$ diameter of rod. In practice it is made a little larger than this, namely from d to $1 \cdot 1\, d$, so as to get a good bearing surface.

(iii.) Considering sections of fork at $a c$, $a^1 c^1$, rod B (Plate XI.) Let d_1 = diameter of pin; D = external diameter $= 2\, d_1$; f = tensile strength of material; width of eye $= t$;

$$\text{then area of the two sections} = (D - d_1)\, t \times 2$$

$$(\text{or since } D = 2\, d_1) = d_1\, t \times 2$$

Equating the resistance of this section to resistance of rod, and taking $d_1 = d$, we have

$$2\, dt \times f = \cdot 78\, d^2 \times f$$

$$2\, t = \cdot 78\, d$$

$$t = \cdot 39\, d,$$

that is, for equal area of section to resist fracture in eye and rod, width of eye $= \cdot 39$ diameter of pin or rod. In practice t is often taken greater than this.

(iv.) Considering section of eye in rod A through $c\, c^1$. Area of section to resist fracture should equal sum of sections through eyes in fork end of B.

The length of the bearing surface of the pin is increased when there is much motion in the joint.

THE DOUBLE NUT OR UNION JOINT (Plate XI.) consists of one long solid nut, into which are screwed two bolts, one at each end. One bolt has a right-handed thread cut on it, and the other a left-handed thread, the effect of which is that when the nut is turned in one direction it draws the two rods together, and shortens the distance between them, and *vice versâ*. The rods are, of course, prevented from turning with the nut. This is a convenient method of adjusting a rod as to length.

KEYS.—Wheels, pulleys, couplings, &c., are prevented from turning round on the shaft by means of flat, wedge-shaped pieces called keys, which are driven between the shaft and the wheel in a groove cut for the purpose. The width of the key is the same throughout, but its thickness tapers, or decreases gradually towards the end at the rate of from about 1 in 64 to 1 in 96. Keys are of three kinds, viz., the saddle key, the key with flat on shaft, and the sunk key.

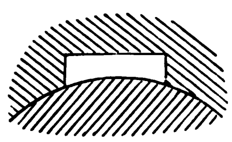


Fig. 32.
Saddle Key.

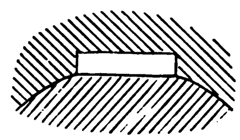


Fig. 33.
Key on Flat.

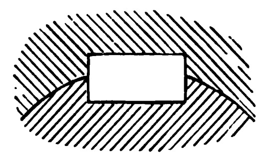


Fig. 34.
Sunk Key.

The "SADDLE," or "HOLLOW," KEY (Fig. 32) is made with its under face concave to fit the surface of the shaft. It is driven into a groove, or key-way, cut in the wheel, and when driven up tightly causes the wheel to grip the shaft. It is only used when the power to be transmitted is small, as in belt pulleys. The advantage of the saddle key is that the pulley can be removed along the shaft and readily secured in any position.

The "FLAT" KEY (Fig. 33) differs from the saddle key in requiring a flat bed to be made for it on the shaft. It is much more common, and a somewhat stronger arrangement than the saddle key.

The "SUNK" KEY (Fig. 34) is the most trustworthy form, and is the one adopted for all purposes where the power to be transmitted is at all considerable. For this key a groove is cut in the *shaft*, in which the key is firmly embedded, the remainder of the key fitting into a corresponding recess in the wheel, as shown in the figure.

FITTING OF KEYS.—All keys are made to fit accurately at the top and bottom, so as when driven up tightly to cause the wheel to grip the shaft. They are also made a snug fit at the sides.

PROPORTIONS OF KEYS.—Width of key = $\frac{1}{4}$ dia of shaft + $\frac{1}{8}$ in.

Mean depth of key = $\frac{1}{2}$ width (for sunk keys).

16. PEDESTALS. PLATE XII.

Pedestals, or Plummer Blocks as they are sometimes called, are used to support rotating shafts. The simplest form of bearing is merely a cylindrical hole in the frame supporting the shafting, but when such a hole wears oval there is no means of readjusting it. If, however, the hole in which the shaft rotates be lined with a cylindrical ring of metal B (Fig. 35) which may take the wear, then when necessary the lining can be easily renewed. These linings are usually made of brass or other soft metal, and are technically called "bushes," or "liners."

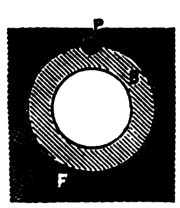


Fig. 35.
F frame, B bush, P pin to prevent bush turning round with shaft.

When the bush, instead of being a continuous ring, is made in two semicircular parts (usually with an overlapping flange at each end), the halves are known as "brasses" or "steps."

This method of making the bush in two parts is the more usual and convenient arrangement, for when the hole wears slightly oval (a result which is followed by a knocking or thumping of the

working parts), the brasses can be readily adjusted, or moved closer together by removing with a file some of the material along the sides, where the two halves are in contact, thus enabling them to be brought together so as again to fit the shaft accurately when tightened up. The example shown in the Plate, and in Figs. 36 and 37, is a type of pedestal used for large engine main

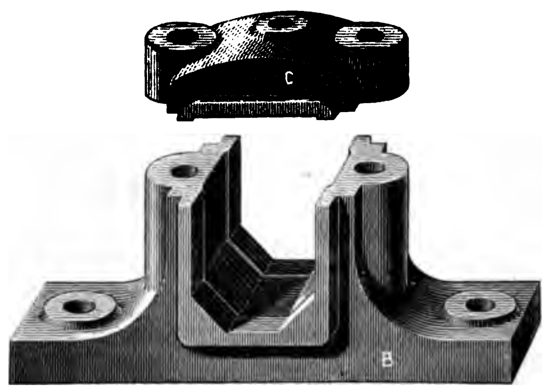


Fig. 36.

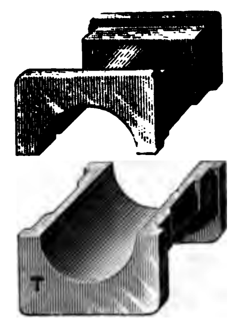


Fig. 37.

bearings, and in this instance is made to carry an 8 in. journal. It consists of the body B, the cap C, the brasses S and T, and the holding-down bolts for cap and base.

On the bottom of the casting a projecting rim of material called the "chipping strip" is cast. This strip is planed up true, so that the pedestal may rest firmly on its foundation. The centre of the casting is shaped to take the lower brass T, and the lower part of the cap is made to fit the upper brass S. The shape of these parts varies according to the shape of the brasses.

The two brasses S and T (Fig 37) are shown removed from their place in the casting. Comparing them with the bush in Fig. 35, it will be noticed that instead of a pin to prevent them from turning with the shaft, there is an octagonal shoulder on the lower half and a square shoulder on the upper half. These shoulders do not extend the whole length of the brass, but only a short distance behind the square flanges shown at each end. The flanges are used to prevent lateral movement of the brasses.

The bolts holding the cap pass right through the casting, terminating at the bottom in square heads, which fit into a square recess, formed in the casting to receive the head, and to prevent the bolt from turning round when the nuts are being screwed up.

The greatest care is necessary in order to properly lubricate bearings. The lubricant, when properly distributed over the bearing, interposes itself as a layer between the rubbing surfaces, and thereby prevents actual contact between metal and metal, thus decreasing friction and preventing undue wear of the parts. When, from any cause, a bearing is not properly lubricated, the metallic surfaces come in contact, and shaft and bearing soon become very hot. Serious breakdowns have frequently resulted from this cause.

The lubricator shown in the plate is an ordinary siphon lubricator, and is made of brass. The tube in the centre of the cup is intended to rise above the surface of the oil surrounding it. One end of a cotton wick is pushed down the tube, the other end overhangs the top of the tube and dips into the oil in the cup. The oil is thus slowly siphoned from the cup to the bearing.

Grooves, not shown in the drawing, are cut on the interior surface of the brasses, to allow the oil to find its way over the whole bearing.

THE PRESSURE PER SQUARE INCH ON BEARINGS $= \dfrac{\text{total load on bearing}}{\text{length of bearing} \times \text{diameter.}}$

In slow-moving machinery 500 lbs. per square inch, or even more, has been found to work satisfactorily, but in quick-running engines the pressures allowed on the lubricant should not exceed 350 lbs. per sq. in.

The maximum load on the bearing shown in the plate may be $12\frac{1}{4} \times 8 \times 500 = 49{,}000$ lbs.

PROPORTIONS OF PEDESTALS.

Unit = diameter of bearing + ½ inch.
Height to centre of bearing = 1.
Distance between centres of cap-bolts = 1·65.
Diameter of cap-bolts = ·25.
Thickness of cap = ·35.
Thickness of metal round cap-bolts = ·3.
Distance between centres of base-bolts = 3·25.
Diameter of base-bolts = same as cap-bolts.
Length of base = 4.
Breadth of base = length of bearing — twice thickness of flange.
Thickness of base outside = thickness of cap.
Length of bearing = 1·25 (increased for high speeds).
Thickness of brass in bearing at top and bottom = ⅛ diameter of bearing + ⅛ in.
Thickness of brass at side = ¾ thickness at bottom.
Thickness of flange = thickness of brass at bottom.
Diameter of flange = diameter of bearing + 3½ times thickness of brass at bottom.

Ex. Design a pedestal for a 2½ ins. shaft, similar to the example given in the Plate XII., but with an octagonal shoulder for the top brass as well as for the bottom. See bearings on engine bed, Plate XXIII. Draw full size. Remember that the *unit* will be 2½ + ½ = 3 ins.

17. BELT PULLEYS. PLATE XIII.

The pulleys shown in the Plate are cast-iron pulleys with curved arms. The arms are often made straight, but unless a pulley is well proportioned, the straight arms are liable to snap on the cooling of the casting owing to the unequal cooling and contraction of the parts, by which internal stresses are set up in the pulley. If fracture does not actually take place, the internal stresses may still exist, at least for some time, and when aided by external forces, result in a breakdown. The proportions shown on the drawing give a good form of arm.

TO DRAW THE PULLEYS.—First draw the circles representing the boss and rim. In Fig. 1 the centre line of the arms is struck with a radius R = the radius of the internal edge of the rim. From point *a*, with radius R, step round the rim, and where these arcs intersect the internal edge we have the centres for the centre lines of the arms. On each centre line of arm set off, by small arcs of circles, as shown in the figure, the thickness of the arm just beyond the boss and within

the rim. It will also be convenient to take the mean of these two dimensions (namely, $\frac{1}{2}$ of $1\frac{7}{8} + 1\frac{1}{8}$) and set it off in the middle of one of the arms, as at *b d*. Now find a radius *b c*, which will describe an arc touching these three arcs, and with this radius draw the convex edge of each arm. Similarly find radius *c d*, which will describe the concave parts of the arms. Round up the points of junction of the arms with the rim and boss.

The arms of Fig. 2 are drawn by first marking off on he centre line E F, from the centre of the wheel, a distance *c s* = one-fifth of the diameter (in this case, $\frac{1}{5}$ of $22\frac{1}{2}$ ins. $= 4\frac{1}{2}$ ins.) From *s* raise an indefinite perpendicular. From centre *s*, with radius *s c*, draw the arc *c a*, cutting the perpendicular in *a*. From *a*, measure *a d* = one-seventh of the diameter (in this case, $\frac{1}{7}$ of $22\frac{1}{2}$ ins. $= 3\frac{1}{4}$ ins. nearly); and from centre *d*, radius *d a*, complete the centre line of the arm. The centre lines of the remaining arms may be drawn by setting off lines corresponding to E F (in this case six) at equal distances round the circle, and working from them as before. Now set off the thickness of the arm on each side of the centre line by small arcs at the top and bottom of the arm; also from *a*, set off the mean thickness of the arm, and complete the arcs as shown. Repeat this method for each arm.

A convenient method of drawing the elliptical section of the arm, when the minor axis of the ellipse is half the major axis, is shown in Fig. 3. Describe a circle whose diameter is equal to the major axis of the ellipse; that is, equal to the breadth of the arm at the given section. The ellipse may now be constructed by two arcs of circles, thus:—Draw an arc from *m* with radius *m n* equal to three-fourths B; repeat this from the other side, and round off the ends by arcs touching the three given arcs.

VELOCITY RATIO OF PULLEYS.—The number of revolutions made by a pair of pulleys connected by a belt is inversely as their diameters—supposing there is no slipping. Thus, if the motion of a pulley 12 in. dia^r is transmitted to a pulley 6 in. dia^r by means of a belt, the small pulley makes two revolutions to one of the large one; or if D and N are respectively the diameter and number of revolutions of the large pulley, and similarly *d* and *n* those of the small pulley, then

$$\frac{D}{d} = \frac{n}{N}$$

The velocity of the *belt* may be taken as the velocity of the rim of the pulley $= D \times 3{\cdot}1416 \times N$, where D = the outside diameter of the pulley in feet. More correctly D should be taken = diameter of pulley + thickness of belt.

Ex. Four pulleys connected by belting are arranged as shown in Fig. 38; the driver A makes 100 revolutions per minute; find the number of revolutions made by D in the same time. Diameters of pulleys = A 3 ft.; B 1 ft.; C 2 ft. 6 ins.; D 10 ins.

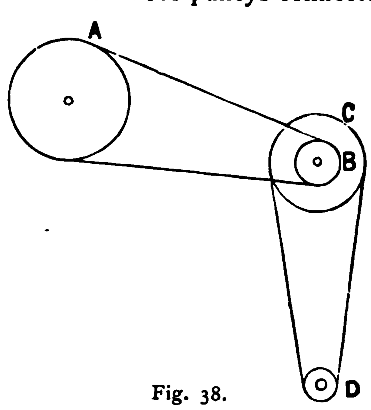


Fig. 38.

$$\text{Revo}^{\text{s.}}\text{ of D} = \frac{\text{Product of Drivers}}{\text{Product of Driven}} \times \text{Revo}^{\text{s.}}\text{ of A}$$
$$= \frac{A \times C}{B \times D} \times 100 =$$
$$= \frac{36 \times 30}{12 \times 10} \times 100$$
$$= 900 \text{ revolutions per minute.}$$

As a belt always tends to move towards that part of a coned or tapering pulley where the radius is greatest,

the rims of ordinary belt pulleys are usually made slightly convex, the result of which is that the belt tends to keep in the centre of the pulley instead of slipping off on either side. The amount of convexity allowed is one twenty-fourth the breadth of the rim.

Belt gearing transmits power by virtue of the friction between the belt and the pulley. When the driving pulley (Fig. 39) begins to move, the result of the friction is that tension (T_2) is set up on one side of the belt greater than (T_1) on the other, and when the difference between the tensions ($T_2 - T_1$) is sufficient to overcome the inertia of the parts to be driven, motion of the driven pulley commences. If R = resistance at circumference of driven pulley; and F = friction between belt and pulley;

$$\text{then } T_2 - T_1 = R = F.$$

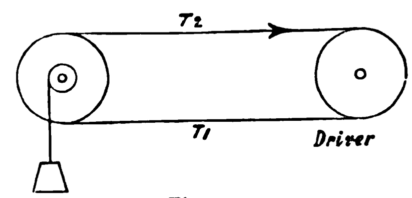


Fig. 39.

The amount of the *friction* between the belt and pulley depends upon (1) the material of which each is composed, different materials having different friction factors. For leather belts on iron pulleys the friction factor = 0·3 to 0·4; (2) the tension in the belt ($T_1 + T_2$); and (3) the length of the angle of contact between belt and pulley.

The horse-power transmitted by belting is equal to the resistance R at the circumference of the driven pulley × velocity of the rim in feet per minute ÷ 33000.

$$HP = \frac{R\,V}{33000};$$

or, if the HP transmitted be known, the resistance may be found thus:—

$$R = \frac{33000\ HP}{V}$$

Ex. 1. An engine transmits 20 horse-power through a belt which passes over a driven pulley 3 ft. in diameter, and drives it at the rate of 150 revolutions per minute; find the resistance at the circumference of the pulley.

By formula $$R = \frac{20 \times 33000}{3 \times 3 \cdot 1416 \times 150} = 467 \text{ lbs.}$$

The power which can be transmitted through a given belt varies with the velocity of the belt; hence, by increasing the diameter of a pair of pulleys while still retaining the same ratio between them, we increase the velocity of the belting, and thereby its capability to transmit power.

Strength of Leather Belts.—Prof. Unwin gives the following formula for the safe working tension f per inch width of belt of thickness t:—

$$f = 320\ t.$$

Thus working strength of belt $\frac{7}{32}$ thick = $320 \times \frac{7}{32}$ = 70 lbs. per inch width.

Ex. 2. Find a suitable width of driving belt in above example where the resistance at the circumference of driven pulley = 467 lbs., and strength of belt allowed = 70 lbs. per inch width.

The belt must be proportioned with reference to the greatest tension, and this is on the tight side of the belt, and is greater than the resistance at the circumference of the pulley, for

$$T_2 - T_1 = R, \text{ as shown above.}$$

Let the ratio between the tensions on the tight and slack sides of the belt be as 2 : 1, which is a useful approximation for ordinary cases in practice.

	Effective pull.		Maximum pull.		Effective pull.		Maximum pull.
Then	$T_2 - T_1$	:	T_2	::	R	:	x
	$2 - 1$	:	2	::	467	:	x

$$x = 934$$

$$\text{Then } \frac{934}{70} = 13{\cdot}3 \text{ ins. width of belt.}$$

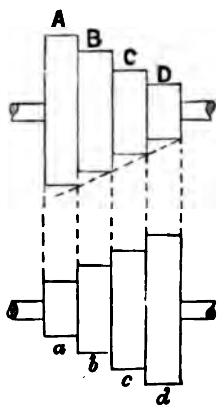


Fig. 40.

Speed Cones.—Speed cones are an arrangement of pulleys of different diameters forming one solid stepped pulley, and arranged to work with a similar pulley placed the reverse way, as in Fig. 40, so that the largest pulley of one works with the smallest pulley of the other.

In order that a crossed belt of constant length may work on any two pulleys of the pair, the ***sum of the diameters of any pair of pulleys must be a constant quantity;*** that is, $A + a = B + b$, &c.

A graphical method of finding the diameters of the intermediate pulleys, having given the diameters of the two end pulleys, is to divide the line in the direction of the axis for the number of pulleys, to draw the first and last pulley as at A and D, and to join them by a straight line A D as shown. The intermediate pulleys must touch this line.

Ex. The pulleys of a speed cone are respectively 16, 13, 10, 7 and 4 ins. diameter, connected by belting with an exactly similar cone; the driving shaft makes 40 revolutions per minute; find the possible speeds of the driven shaft. *Ans.* 160, $74\frac{2}{7}$, 40, $21\frac{7}{13}$, 10.

18. COUPLINGS. Plate XIV.

Flanged Coupling.—The explanatory sketch (Fig. 41 A) shows the flanged coupling taken to pieces. In the plate, the parts are shown all properly connected. Fig. 1 *assumes* the coupling to

Fig. 41 A.

be cut through, as shown by the dotted lines in Fig. 41 A. The upper half of Plate XIV. shows an exterior view or elevation of the face plates FP^1 and FP^2; the lower half shows an interior view of the parts through which the section is assumed to be taken. In Fig. 2, the upper half shows an exterior view of the face plate FP^1, and the lower half shows the inner face of FP^2.

Shaft couplings are used to connect short lengths of shafting so as to make one long length. The flanged or face-plate coupling consists of two cast-iron discs or face plates, each of which is firmly secured on the end of its respective shaft by a key K; the two discs are then bolted together, and the two shafts are thus effectively coupled. It will be observed that the head and nut of each bolt are sunk into the body of the flange. This is a precaution in order to prevent accidents from the bolts entangling the workmen's clothes. The end of one shaft projects forward into the flange of the other to ensure that both shafts shall be in line. On the interior face of the face plate, around the rim, bolt holes, and boss, the metal is seen to project above the rest of the face. This raised part is called "facing strip," and it is faced up perfectly true in a lathe after the face plate has been secured to its shaft, thus ensuring that the plane of the face shall be perpendicular to the axis of the shaft. The recessed parts are left rough.

As your drawing is to be done half full size, take half the given dimensions in each case. Notice that the centres of the six bolts are all on the circle 11¾ ins. diameter, and that the radius divides the circumference into six equal parts.

Half-lap Coupling.—In this coupling (Fig. 41 B) the ends of the shafts are cut away as shown, and the two halves are made to overlap one another. The cylindrical cast-iron box is then placed over them to keep them together. The box is secured by a saddle key. The two halves are

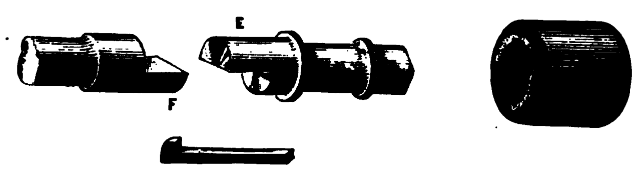


Fig. 41 B.

tapered in such a way as to prevent their slipping apart in the direction of their length. The taper of 1 in. per foot, or one in twelve, is obtained by measuring from the intersection of the two centre lines MN and GH, a length of twelve parts (say inches) to the right on the centre line; from the point thus obtained, raise a perpendicular one part (inch) high; then the line joining the top of the perpendicular with the intersection of MN and GH gives the taper required.

The shafts, bolts, and keys are of wrought-iron, or sometimes steel; the other parts are cast-iron.

A simple form of coupling sometimes adopted, and called a Box Coupling, is one in which the ends of the shaft to be connected merely butt against one another, meeting in the centre of the box, instead of overlapping as shown. The two shafts are keyed to the box by separate keys, or by one long key passing through the whole length of the box.

Coupling Bolts.—In well made couplings, the bolts are subjected to a shearing stress only, through the section of the bolt cut by a plane passing between the flanges.

Allowing safe shearing stress = 8000 lbs. for wrought-iron bolts,

then resistance of each bolt to shearing $= \frac{\pi}{4} d^2 \times 8000$;

and if n = number of bolts, then total resistance to shearing $= n \times \frac{\pi}{4} d^2 \times 8000$;

and, if r = radius of bolt circle, moment of resistance of bolts to shearing

$$= n \times \frac{\pi}{4} d^2 \times 8000 \times r;$$

and this must equal twisting moment of shaft, for bolt and shaft to be equally strong; but twisting moment T, for shafts of diameter D, is given by the formula (page 25).

$$T = \frac{\pi D^3}{16} f;$$

$$\text{taking } f = 8000 \text{ and } r = 0{\cdot}8\ D,$$

$$\text{then } \frac{\pi}{4} d^2 \times n \times 8000 \times {\cdot}8\ D = \frac{\pi D^3}{16} \times 8000;$$

$$\text{from which } d = \frac{D}{2}\sqrt{\frac{1}{{\cdot}8\ n}}.$$

If 4 bolts be used, then

$$\text{diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{{\cdot}8 \times 4}}$$

$$= \frac{D}{3{\cdot}5}.$$

If 5 bolts be used, then

$$\text{diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{{\cdot}8 \times 5}}$$

$$= \frac{D}{4}$$

19. INTERPENETRATION OF SOLIDS. PLATE XV.

The method of obtaining the true curve formed at the junction of the surface of two cylinders is as follows:—Draw a semi-circle on one end of the horizontal cylinder in both plan and elevation. (Fig. 1). Divide each of the semi-circles into any number of equal parts (say eight), and number them 1′, 2′, 3′, &c., 1, 2, 3, &c. Notice that 1 in the plan, is another view of the same point 1′ in the elevation; also 5 and 5′ are two views of one and the same point, and so on.

Draw horizontals through 1, 2, 3, &c., in the plan, and produce them till they meet the circular plan of the vertical cylinder in points *a*, *b*, *c*, *d*, &c. Points *a′*, *b′*, *c′*, *d′*, &c., in the elevation, are now obtained by projecting from *a*, *b*, *c*, &c., in plan, vertical lines to cut horizontals through 1′, 2′, 3′, &c. A curve representing the interpenetration of the two cylinders is then drawn by hand through the intersections thus obtained.

A similar method is adopted to find the interpenetrations in Fig. 2.

20. STEAM COCK. Plate XVI.

When water or steam is passed from one chamber to another through a pipe, it is necessary to have a means of opening, or closing, or regulating the communication between the two chambers. This is done by inserting a fitting similar to that in Plate XVI. Fig. 42 is an explanatory sketch representing the steam cock taken to pieces. The flanged end FE may be bolted to the boiler, and the pipe may be attached to the screwed end SE. The cock consists of a conical plug CP with a hole PH through it, and a hollow casing or shell, which is bored out to exactly fit the plug, and having a passage through it from one side to the side opposite. When the plug is in its place, by turning it in one direction the hole through it will coincide with the hole through the shell, and the communication is then open. By turning it through one-fourth of a revolution, as shown in the Plate, the communication is cut off.

Fig. 42.

The example affords an instructive exercise in projection. The construction showing how to obtain the curve at the junction of the pipes with the shell will be better understood after working the exercise on Interpenetration of Solids. (Plate XV.) The dotted semicircle to the left of the figure represents half of the pipe at its largest diameter, where it is rounded up to meet the shell. The semicircle is divided into any number of equal parts (say four), and horizontal lines are drawn through the divisions towards the shell. These lines meet the shell in points, which, when found, will give the curve of the intersection required. To find them, draw a part plan thus:—From the centre *r*, with radii *ra rb rc*, &c., equal to *r'a'*, &c., describe arcs *a b c*, &c. These arcs represent the circumference of the shell at sections through *a' b' c'*, &c., perpendicular to the axis. Draw part of the semicircle 012, divided as before, and draw horizontals through the divisions to cut the arcs. Then, from the points of intersection of these lines with the arcs *a b c*, &c., raise perpendiculars to intersect the corresponding horizontal lines through *a' b' c'*, &c. A freehand curve through these intersections is the curve required. It can be ruled in ink by using a "French curve."

The reference letters on the plan correspond to similar letters on the sectional elevation. The whole of the parts are of gun metal.

Chapter VI.

ENGINE CONSTRUCTION.

21. CRANKS AND CRANK SHAFTS. PLATES XVII. AND XVIII.

CRANKS are used to convert the reciprocating motion of the piston into the circular motion of the main shaft by means of the connecting rod which takes hold of the crank pin.

Figs. 1 and 2, Plate XVII., are two views of a three-throw shaft. Fig. 1 is drawn first, and the centres of the cranks are set out at angles of 120° with each other. The several edges and corners of the cranks in Fig. 2 are all projected from Fig. 1.

The three-throw crank shaft is forged solid. The journals are those parts of the shaft which rotate in the bearings supporting the shaft. For shafts running at low speeds the journals are often not more than one diameter long; usually the length varies from $1\frac{1}{2}$ to 2 times the diameter, the length increasing with the speed. The advantages of long journals in quick running machinery are now fully recognised.

The corners of journals of shafts should always be rounded; sharp corners or abrupt changes of section considerably reduce the strength of shafts.

Figs. 3 and 4 are two views of a wrought-iron overhung crank, made separate from the shaft.

This crank consists of an arm with a boss at each end—one to take the crank shaft and the other the crank pin. The crank is secured firmly in its place on the shaft either by keying alone, or by " shrinking " and keying. The shrinking is done by boring out the hole a shade smaller than the shaft, then heating the crank round the hole and thus causing the material to expand and the hole to become larger. The crank is then slipped in its place on the shaft, and on cooling it contracts and grips the shaft tightly. Forcing on by hydraulic pressure is now frequently adopted in preference to shrinking. The crank pin is shrunk in position or forced in by hydraulic pressure, and riveted over the end, as shown. Great care is taken in the design and construction of the crank pin, and to properly secure it in position, as the various stresses and shocks to which the engine is subject are felt most severely upon the crank pin.

To show the method of obtaining the correct form of the curves at A and B, Fig. 4, the parts have been enlarged in Figs. 5 and 6: thus A and B in Figs. 3 and 4 correspond with A and B in Figs. 5 and 6.

The curves are formed at the junction of the surfaces which form the sides of the crank arm with that passed over by the lathe tool in turning up the circular portion of the bosses.

Divide the arc $a' m' b'$, Fig. 6, into any number of parts, and project from these divisions by horizontal lines to $a\,m\,b$ on the vertical line CD. From the centres C and D describe arcs cutting the edge AB. From these points of intersection project back horizontals to meet verticals from corresponding points on the arcs $a'\ m'\ b'$. These last intersections when joined give the curve required.

The radius of the crank arm is measured from the centre of the shaft to the centre of the crank pin. The radius is equal to half the throw of the crank. The *throw* of the crank is equal to the diameter of the crank pin path, and is also equal, of course, to the stroke of the piston (except where the motion is not direct, as with the beam engine). The crank pin makes one revolution for a forward and backward stroke of the piston. Hence the piston moves through two strokes while the crank pin moves through 3·1416 times the stroke; or, mean velocity of piston is to mean velocity of crank pin as 2 : 3·1416, or as 1 : 1·5708.

The principal stress which shafts are called upon to bear is the torsional or twisting stress. The strength of shafts to resist torsion is in proportion to the cube of their diameters; thus, if we know the strength of a 1 in. bar to resist torsion, then a 2 in. bar is to a 1 in. bar as 1^3 is to 2^3, that is, as 1 is to 8, or, a 2 in. bar is 8 times as strong as a 1 in. bar, a 3 in. bar is 27 times as strong, and so on. (See Ex. 2, page 25).

The diameter of shafts subjected to twisting stress is determined from the following formula :—

$$d = \sqrt[3]{\frac{T}{f} \times 5{\cdot}1} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \text{ (see page 25)}$$

Where d = diameter of shaft in inches.

T = maximum twisting movement, that is, greatest load on crank pin in lbs. × radius of crank in inches.

f = safe working stress in lbs. = 8000 for wrought-iron ; 12000 for steel.

Ex. 1. Find the diameter of a wrought-iron crank shaft for an engine, considered with reference to twisting, when maximum pressure on the crank pin at half stroke is 7500 lbs. and radius of crank arm 12 ins.

$$d = \sqrt[3]{\frac{7500 \times 12}{8000} \times 5{\cdot}1}$$

$$= \sqrt[3]{57{\cdot}4}$$

3·85, or $3\frac{7}{8}$ ins.

To allow for proper lubrication the dimensions of crank pins and shaft journals are made to depend upon the pressure per square inch on the bearings, measuring the effective bearing area by multiplying the diameter of the bearing by its length.

The pressure allowed in different cases in practice varies very considerably, but according to Mr. Seaton it should not exceed 500 lbs. per square inch for crank pins and shaft journals.

Ex. 2. The diameter of a cylinder is 20 ins. and the steam pressure at commencement of stroke is 80 lbs.; supposing the pressure allowed on the bearing surface of the crank pin to be 500 lbs. per sq. in., find the dimensions of the pin when its length is $1\frac{1}{4}$ times its diameter.

Then,
$$\text{Length of pin} \times \text{diameter} = \frac{\text{total load}}{500}$$
$$\text{that is,}\quad 1\tfrac{1}{4}\ \text{diameter}^2 = \frac{20^2 \times \cdot 7854 \times 80}{500}$$
$$\text{diameter} = \sqrt{40}\ \text{nearly}$$
$$\text{diameter} = 6\tfrac{3}{8}\ \text{ins.}$$
$$\text{Length of pin in bearing} = 6\tfrac{3}{8} \times 1\tfrac{1}{4} = 8\ \text{ins.}$$

Ex. 3. Considering the pin in the above Example (2) to resist bending. Suppose the crank is a single arm, or overhanging crank, as in Fig. 43.

Then, moment of resistance = bending moment
$$\frac{f\,d^3}{10\cdot 2} = \text{P} \times l \quad \dots \quad \dots \quad \dots \quad \text{(see Table VI., page 21)}$$
Where $f = 8000$ for wrought-iron, d = diameter of pin, P = thrust of connecting rod when crank is on the dead centre = 25,000 lbs., l = distance between centre line of engine and face of crank = 5 ins.

Then,
$$d = \sqrt[3]{\frac{\text{P}\,l}{f} \times 10.2}$$
$$d = \sqrt[3]{\frac{25000 \times 5}{8000} \times 10.2}$$
$$= \sqrt[3]{160}\ \text{nearly}$$
$$= 5\cdot 43\ \text{inches}$$
Showing that if the dimensions of the pin were chosen so as to provide for not more than 500 lbs. pressure on the bearing surface of the pin, it is amply strong to resist bending. (See Example 2.)

Considering the bending action on the crank pin for a crank as in Fig. 44, midway between two bearings. If l = length between centres of bearings, and P = load on crank pin; then, when crank is on the dead centre,
$$\text{Bending moment on pin} = \frac{\text{P} \times l}{4} \quad \dots \quad \dots \quad \text{(see Table V., page 20)}$$
When a crank pin transmits HP (horses' power), the mean pressure p on the pin is obtained from the equation:
$$p \times 2\pi r n = 33000\ \text{HP}$$
$$\text{or } p = \frac{33000\ \text{HP}}{2\pi r n} \quad \dots \quad \dots \quad \dots \quad \dots \quad \dots \quad (1)$$
where $2\pi r$ = the length of the circular path of crank pin of radius r, in feet, and n = number of revolutions per minute.

From this equation we may also obtain the mean twisting moment.
$$p\,r = \frac{33000\ \text{HP}}{2\pi n}\ \text{foot lbs.}$$
$$= \frac{33000\ \text{HP} \times 12}{2\pi n}\ \text{inch lbs.}$$
$$= \frac{63025\ \text{HP}}{n}\ \text{inch lbs.} \quad \dots \quad \dots \quad \dots \quad \dots \quad (2)$$

Crank shafts, being supported between bearings, are subjected to both a twisting and bending action at the same time, and hence to determine correctly the diameter of a shaft at a given section, we must consider the combined effect of these straining actions at that section (page 26).

The amount of straining action on the crank shaft varies at different parts of its length. Where there are two or more cranks to one shaft, the crank nearest the resistance (which in marine engines is the crank nearest the propeller or the "after" crank) is subjected to the greatest straining action. The forward part has to transmit the power of the forward engine only, but the after part the power of all the engines, so that different dimensions might be employed for different parts of the shaft, the dimensions increasing towards the resistance. For convenience, however, and to make the parts interchangeable, the cranks and shaft are made of the same dimensions throughout.

Ex. 4. In the single arm crank (Fig. 43), if the distance m, namely, the distance between the centre line of the engine and the centre of the bearing, be 15 ins., and the radius of the crank arm be 15 ins.; find the diameter of the shaft journal necessary to resist the combined bending and twisting stresses.

Bending moment $= P \times m$.

Mean twisting moment $= P \times r \times \cdot 75$ approximately.

As before, let $P = 25000$ lbs.

Equivalent bending moment $M^1 = \cdot 9\,M + \cdot 4\,T$ (page 26);

and moment of resistance = bending moment;

$$\text{therefore } \frac{f\,d^3}{10 \cdot 2} = \cdot 9\,P\,m + \cdot 4\,P\,r \times \cdot 75$$

$$= P\,m\,(\cdot 9 + \cdot 3) \text{ (since } P\,m = P\,r)$$

$$\text{or, } d = \sqrt[3]{\frac{P\,m \times 1 \cdot 2}{f} \times 10 \cdot 2}$$

$$= 8 \cdot 3 \text{ inches.}$$

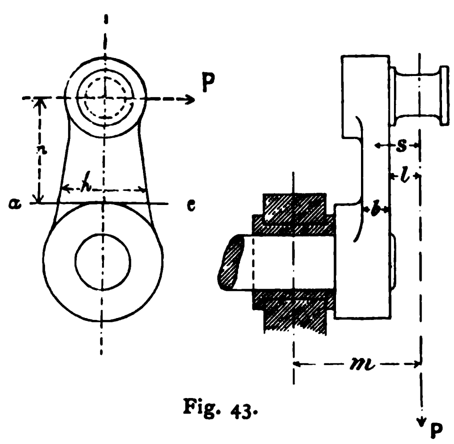


Fig. 43.

A common rule for the diameter of engine shafts and crank pins is as follows:—

$$\text{Diameter} = 4 \cdot 5 \sqrt[3]{\frac{\text{Indicated horse-power.}}{\text{Revolutions per min.}}}$$

For shafting subjected to twisting only:—

$$\text{Diameter} = 4 \sqrt[3]{\frac{\text{Indicated horse-power.}}{\text{Revolutions per min.}}}$$

Crank Arms. The dimensions of the rectangular arm of an overhanging crank are determined from the following considerations. Referring again to Fig. 43:—

1. When the engine is on the dead centre, the pull or thrust P on the crank pin causes a combined straining action on the crank arm, consisting of a direct tension or compression P, and a bending moment $P\,s$ (page 23); tending to cause fracture at the smallest section of the arm close to the crank pin boss.

2. When the crank arm and connecting rod are at right angles, the force P acting on the pin causes a combined straining action at any section $a\ c$ of the arm, consisting of a bending moment $P\,n$, and a twisting moment $P\,s$, and tending to cause fracture at the root of the arm close to the crank shaft boss.

Considering the first case; by page 23 the resistance of the section to bending when under the action of a combined pull or push and bending moment is expressed by the formula $\left(f - \frac{P}{a}\right) z$; and equating this to the bending moment,

$$\left(f - \frac{P}{a}\right) z = Ps$$

$$\left(f - \frac{P}{bh}\right) \frac{bh^2}{6} = Ps$$

$$\text{or, } f = \frac{P}{bh}\left(1 + \frac{6s}{h}\right)$$

where a = area of section of arm; z = modulus of rectangular sections for bending $= \frac{bh^2}{6}$; f = 8000 for wrought-iron.

Assuming a value for h, or the width of the arm at the smallest section (say 1·5 diameter of crank pin), we can obtain from the above equation the value of the thickness b.

Considering the straining actions in the arm for the second case, when the crank and connecting rod are at right angles.

The combined bending moment Pn and twisting moment Ps are expressed as an equivalent bending moment M^1, thus :—

$$M^1 = \cdot 9\,M + \cdot 4\,T$$

where M = bending moment and T = twisting moment. (See page 26.)

Then, equating this to the moment of resistance of the arm to bending, we have—

$$\frac{f\,b\,h^2}{6} = \cdot 9\,M + \cdot 4\,T$$

Assuming a value for h, the thickness b of the arm can be found at once. The thickness b of the crank arm is usually the same throughout, and the larger of the two values for b, obtained by the above equations, is the thickness selected.

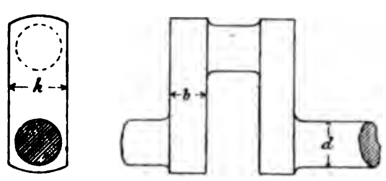


Fig. 44.

The dimensions of the rectangular arms of cranks, as in Fig. 44, are found by making the width h of the arm on the plane at right angles to the axis = 1·2 the diameter (d) of the shaft, and the breadth b = 0·8 d, or such that $b\,h^2 = d^3$.

22. CONNECTING ROD. Plate XIX.

The Connecting Rod connects the cross-head at the end of the piston rod with the crank pin, and by its means the reciprocating motion of the one is transformed into the circular motion of the other. Connecting rods are of various kinds; the one shown in the plate, and further illustrated in Fig. 45, is known as the marine type.

The rod is circular in section, terminating in a T shape at one end and a fork at the other, and slightly bulging in the middle. A steel pin, called a gudgeon or cross-head pin, passes through the fork end, and secures the rod to the cross-head. The gudgeon is prevented from turning round in

its place by a small steel stop, which enters partly into the head and partly into the fork. There is also a shoulder on the pin, the diameter decreasing from $1\frac{1}{4}$ in. to $1\frac{1}{8}$ in. This prevents the ends of the fork from being forced together when the nut is screwed up tight.

Fig. 45.

The crank end of the rod is made up of the T end, two half brasses, and a wrought-iron cap, all of which are bolted together as shown.

To obtain the curves formed by the section of the plane of the face with the curved part of the rod under the T end, first mark off on each view the diameter of the rod $1\frac{1}{8}$ in. Draw the curve $a' \, b' \, c'$ with radius of 1 in. from centre m to touch the rod and head. Before completing the curves, Fig. 2 must now be drawn. Find the centre n by projecting across from m and taking a point 1 in. (same radius as from m) from the rod. Draw the arc, cutting the face in e. Now project from e to the centre line AE of Fig. 1, and thus obtain e', the bottom point of the curve. The remainder can now be approximately completed by arcs of circles. A more exact method of completing the curve is to take any points a', b' and c' in it, and from centre p in plan of cap, cut the plan of the face in a, b and c with radii pa, pb, pc, equal respectively to the distance of a', b' and c' from the centre line. Verticals from these points to cut horizontals through $a' \, b' \, c'$ will give, at their intersections, points in the required curve.

A similar method is adopted to obtain the curves at the fork end of the rod. The dotted figure on line $s \, t$ shows a partial section through the fork, and it is drawn to obtain the position of the corner p, which, with t, gives the double line shown round the edge.

The length of the connecting rod is measured from the centre of the crank pin bearing to the centre of the gudgeon. It will be noticed that the rod is shown broken in the middle; this is usual in practice when very long uniform rods are required, as all that is necessary is to give their diameter and total length. The dimension (1 ft. 6 ins.) informs the workman how long the rod is between the centres.

Connecting rods vary in length from two to three times the length of stroke. They should if possible never be less than twice the length of stroke.

The following figure shows how to obtain the principal centres in striking out a complete engine :—

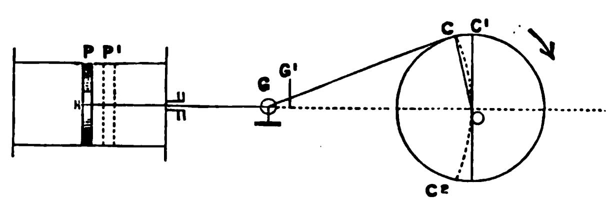


Fig. 46.

P piston. PG piston rod. G cross-head. GC connecting rod. CO crank arm.

Let the piston P (Fig. 46) be in the centre of its stroke; then the gudgeon or cross-head pin at G will also be in the centre of its stroke. Set off from G to O a distance equal to the length of the connecting rod. Then O is the centre of the crank shaft. From centre O, with radius equal to the crank arm OC (or half the stroke of the piston), describe the circle CC^1C^2. This is the crank pin path. From centre G, with radius GO, describe the arc OC, cutting the crank pin path in C.

This is the position of the crank pin when P and G are in the middle of their stroke. It will now be seen that C and C^1 do not coincide. When C reaches C^1, then P and G will have moved forward beyond their middle position to P^1 and G^1. With the piston in its middle position, C would coincide with C^1 if the connecting rod were infinitely long.

The diameter of connecting rods may be obtained from the following formula:—

$$\text{Diameter at centre} = \frac{D}{60}\sqrt{p}$$

$$\text{Diameter at neck} = \frac{D}{65}\sqrt{p}$$

where D = diameter of cylinder in inches;
p = absolute initial steam pressure in lbs. per sq. in.

Connecting rods are sometimes made to taper uniformly from the cross-head end, where it is smallest, to the crank end.

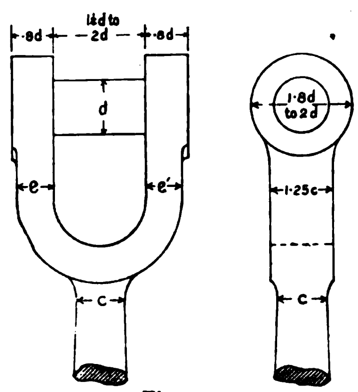


Fig. 47.

Proportions for fork end of connecting rods, Fig. 47:—Area of sections through e and e' should be from 1¼ to 1½ times area of rod at c.

The stress on connecting rod bolts of the marine type of head is a tensile one, but owing to the possibility of shocks being brought upon the bolts on account of slackness of brasses the maximum stress in lbs. per sq. in. allowed is 6,000 for wrought-iron and 8,000 for steel. The total load on the bolts equals the load on the piston; hence each bolt is designed to carry half the load on the piston at its weakest section, namely, at the bottom of the thread.

Then, area of bolt at bottom of thread × 6,000 = area of piston × max. effective pressure of steam ÷ 2.

Taking area at bottom of thread = ·8 area of shank of diameter d; D = diameter of cylinder; p = maximum effective pressure of steam in lbs. per sq. in.

$$\cdot 8\, d^2 \times \frac{\pi}{4} \times 6000 = \tfrac{1}{2}\, D^2 \times \frac{\pi}{4} \times p$$

$$d^2 = \frac{D^2 p}{\cdot 8 \times 12000}$$

$$d = \frac{D}{98}\sqrt{p} \text{ for wrought-iron.}$$

or, for steel

$$d = \frac{D}{113}\sqrt{p}$$

The bolts are placed as close as possible to the crank pin.

The thickness of the brass between crank pin and cap = ⅛ diameter of pin + ¼ in.

The cap is in the condition of a beam supported at both ends and loaded uniformly. Its thickness may be made equal to half diameter of rod.

23. PISTON ROD AND CROSS-HEAD. PLATE XX.

In the example given in Plate XX., the piston rod is secured to the piston, on the one side by a nut, which is prevented from slacking back by a small pin passing right through nut and rod; and on the other side by the tapering of the rod, and the addition of a collar solid with the rod.

The rod is fastened to the cross-head by a tapered steel cotter, A, Plate XX. The clearance, or draft, to allow of the cotter being driven further in, should be noticed. The round hole in the cross-head takes the pin or gudgeon by which the connecting rod is attached (see coloured general drawing). This pin is rigidly secured to the connecting rod; hence the wear caused by the motion of the connecting rod relatively to the piston rod is all felt in the cross-head brasses C. The exact form of the cross-head, and the means of tightening up the two semicircular brasses which form the bush, are shown in the wood-cut, Fig. 48, and in the isometric view on the Plate. (This view is merely explanatory, and is not intended to be drawn by the student.) The cross-head is, in this case, made of gun metal. It is sometimes made of cast-iron or cast-steel.

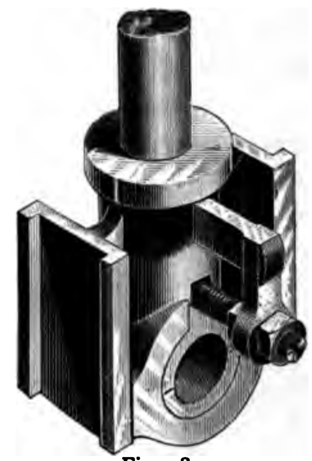


Fig. 48.

PISTON RODS are subjected to alternate pushing and pulling stresses, which occur in rapid succession, and which must severely test the material of the rod; and they are now almost universally made of steel. The weakest part of the piston rod is at the screwed end which takes the nut. This part, however, is only subjected to tension, and not to alternate tension and compression, as the student will see on consideration. When the steam enters the cylinder underneath the piston (see Plate XXIV.), the whole load is carried by the screwed part of the piston rod; but on the return stroke, when the piston is descending, the stress is removed from the screwed part, and comes on the tapered part of the rod and the collar.

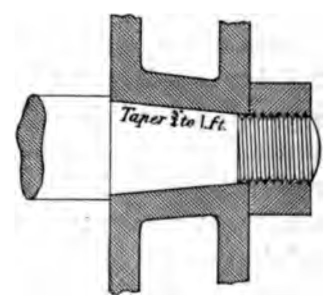


Fig. 49.

The load to be carried by the piston rod equals the difference between the pressure on the two sides of the piston multiplied by the area of the piston (neglecting area of rod). Thus, in a condensing engine, the pressure on one side of the piston equals the boiler pressure by gauge, plus 15 lbs. pressure of atmosphere, minus loss of pressure between boiler and cylinder, minus back pressure due to imperfect vacuum in condenser.

The following formula gives the diameter of piston rods for ordinary cases:—

$$\text{Diameter of piston rod} = \frac{\text{Diameter of Cylinder}}{60} \times \sqrt{p},$$

where p = maximum effective pressure of steam per sq. in.

Ex. Find the diameter of piston rod for a 25 ins. cylinder; maximum pressure = 100 lbs.

In locomotives the diameter of the piston rod is about one-sixth the diameter of the cylinder.

The diameter of the rod in its weakest part, namely, at the bottom of the thread, must be so designed that the stress per sq. in. does not exceed 6,000 lbs. for iron, and 8,000 for steel. In

compound and triple expansion engines it is usual to retain the same dimensions for the piston rods, cross-heads, and connecting rods throughout.

PISTON ROD BOLTS.—When the cross-head end of the piston rod is fitted, as shown in Plate XXI., the two bolts which secure the piston rod, cap, and brasses, should be of such a size that the stress per sq. in. at bottom of thread does not exceed 6,000 for iron, or 8,000 for steel. In other words, the sum of the areas of these bolts at their weakest section should not be less than the screwed end of the piston rod at its weakest section.

CROSS-HEADS. Plates XX. and XXI.—The cross-head consists of a block into which the piston rod is inserted and secured, of a pin or gudgeon passing through the block, upon which the connecting rod works, and of a foot or shoe, which works between guide bars parallel to the axis of the cylinder.

The block and guides prevent the oblique thrust or pull of the connecting rod from bending the piston rod. This can be seen by reference to Fig. 50. When the piston P is being impelled forward so that the rotation of the crank pin is in the direction of the arrows, the resistance at the crank pin c causes a downward thrust R through the connecting rod gc, which may be resolved into two forces, one tending to compress the piston rod, and the other to bend it in the direction g T, causing a downward thrust upon the guides. Again, when the piston is being driven back by the steam, the resistance at the crank pin at c' causes a downward pull through the connecting rod, and the tendency is still to bend the piston rod downwards, and to cause, therefore, a downward thrust upon the guides. If the engine were reversed, the whole of the conditions would be reversed, and the thrust g T would be upwards instead of downwards. Hence the prevailing direction in which horizontal engines should run is that shown by the arrow in the figure, so that the pressure on the guides shall be upon the lower, rather than the upper, guide bar. This is especially important for the sake of efficient lubrication.

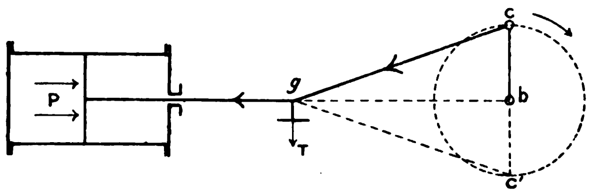


Fig. 50.

It should be noticed that when the crank drags the piston, as it does under some circumstances, the direction of the thrust upon the guides is reversed; hence the necessity for a top and bottom guide bar under all circumstances. The amount of the thrust upon the guide varies from zero at each end of the stroke to its greatest value when the crank is at right angles to the centre line of the cylinder (assuming that the steam is not cut off before this point is reached).

In designing a guide block it is important that the downward pressure through the block should not exceed a certain value, and it is therefore necessary to approximately determine beforehand the greatest value of the thrust for the given conditions.

In Fig. 50, if P g b be the centre line of the engine, c b the crank arm at right angles to it, and c g the connecting rod, drawn to scale; and if the total pressure P on the piston be represented by the line g b, then the length of the line c b to the same scale is the measure of the downward thrust T; or

$$P : T :: gb : cb$$

$$\text{then } T = \frac{P \times cb}{gb} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (1)$$

If the length of the connecting rod be taken instead of $g\ b$, which is sufficiently near for practical purposes, we have from (1)

$$\text{Maximum thrust on guides} = \frac{\text{load on piston} \times \text{radius of crank in ins.}}{\text{length of connecting rod in ins.}}$$

It is evident that the shorter the connecting rod compared with the crank, the greater its obliquity at the centre of the stroke and the greater the vertical thrust T.

Ex. Find the thrust on the guides from the following data :— Load on piston = 20000 lbs. radius of crank 15 ins.; length of connecting rod = 5ft. From above equation,

$$\text{Thrust} = \frac{20000 \times 15}{60} = 5000 \text{ lbs.}$$

The pressure per square inch allowed on the guides varies considerably in different engines; experience teaches that from 40 to 50 lbs. per square inch maximum pressure may be allowed where cast-iron motion blocks efficiently lubricated move on cast-iron slides, without causing any appreciable wear. Hence in the above example the area of the base of the guide block or slipper should be $5000 \div 50 = 100$ sq. ins,; or, say, $8\frac{1}{2} \times 12$ ins.

In marine engines from 70 to 100 lbs. per sq. in. maximum pressure is allowed.

THE GUDGEON OR CROSS-HEAD PIN is so proportioned that the maximum pressure upon it does not exceed 1,200 lbs. per sq. in., the effective bearing area being obtained by multiplying its length by its diameter. With a greater pressure than this it is found in practice that the lubricant is squeezed out. The diameter of the gudgeon is usually $1\frac{1}{4}$ times the diameter of the piston rod, and its length measured in the bearing is from $1\frac{1}{4}$ to twice its own diameter.

Ex. 1. An engine with a cylinder 10 ins. diameter, working at a maximum pressure of 100 lbs. per sq. in., has a cross-head pin $2\frac{1}{4}$ ins. diameter and $4\frac{1}{2}$ ins. long; find the maximum pressure per sq. in. on the bearing area of the pin.

$$\text{Pressure per sq. in. on pin} = \frac{\text{total load}}{\text{length} \times \text{diameter}}$$

$$= \frac{10 \times 10 \times \cdot 7854 \times 100}{2\cdot 25 \times 4\cdot 5}$$

$$= 776 \text{ lbs.}$$

24. STANDARDS. PLATE XXII.

This drawing requires little explanation. Some of the most important dimensions required for laying down the General Drawing are given on this Plate. The position of the standards (which are four in number) will be at once understood by referring to the Coloured Plates XXXII. and XXXIII. They are bolted to the bed plate at the foot, and at the top they support the cylinders. They are also used to carry the weigh shaft and brackets for the starting gear.

Wrought-iron, or steel, turned pillars are sometimes substituted for cast iron-standards.

25. THE BED-PLATE. PLATE XXIII.

The bed-plate, or foundation of the engine, is a piece of cast-iron frame-work which carries the main bearings for the crank shaft. It is also supplied with facing strips on the top to take the feet of the standards and its own holding-down bolts; and on one end there are facing strips to take the feet of the feed and bilge pumps.

The Fig. 51 will show the student the exact shape of this casting.

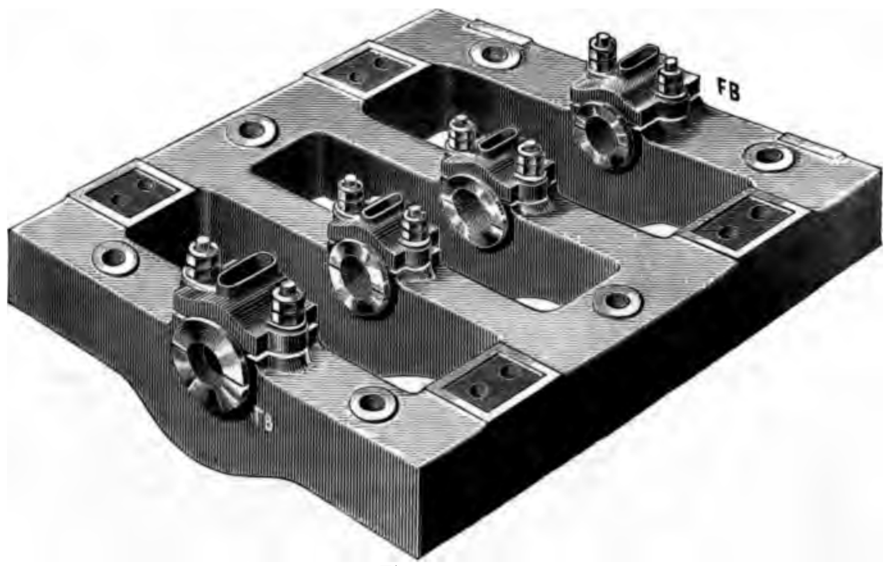


Fig. 51.

26. CYLINDERS. PLATE XXIV.

The Illustration, Fig. 52, shows an isometric view of the cylinder for the launch engines, lying on its side. The slide valve is removed, and the valve box, with its valve face and steam and exhaust ports, is exposed to view.

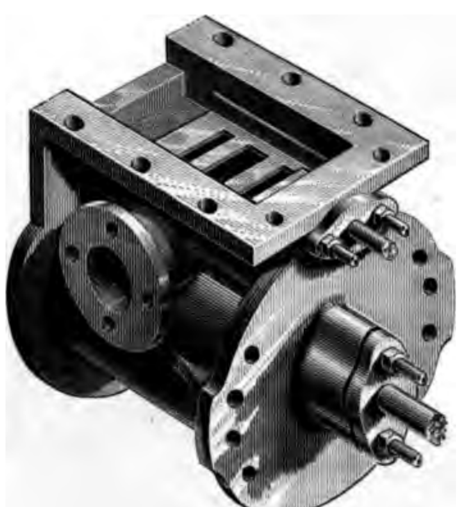


Fig. 52.

The rectangular flange, shown with its face upwards and extending round three sides of the valve box, fits against and is bolted to an exactly similar flange on the other cylinder of the pair, and the joint is made perfectly steam-tight.

It will be noticed that even when the cylinders are bolted together there is still an opening into the valve box at the upper end. This is to admit of the slide valves being placed in position on their respective valve faces. When the valves are in their places the opening is covered up by a flat cast-iron plate, called the valve box cover VBC, which is bolted to flanges on the cylinder. The steam supply from the boiler passes through a pipe $1\frac{1}{2}$ in. diameter to the valve box through a hole in the cover.

The Plate shows plan of top of one cylinder, with a sectional plan of the other in position, with the valve box flanges face to face. One half of the steam pipe and valve box cover is shown.

A longitudinal section of one cylinder is given; while the other, instead of being shown bolted to it, is rotated on its axis, so as to give a view of the valve face and ports.

The cylinder is bored out perfectly true in a lathe, and the piston is turned to fit the cylinder, so that it will move freely from end to end of the cylinder. The piston, though a comparatively loose fit, is made perfectly steam-tight by using spring rings, which are fitted into grooves in the piston, and which press outwards against the cylinder.

The section through the centre of the piston shows it to be hollow. The two-screwed plugs are used to plug up the holes which have been made in order to get out the core of sand or loam from the inside of the piston.

The slide valve SV, and the steam and exhaust ports SP, EP, SP, are an ingenious arrangement for giving to the piston a continuous to-and-fro motion.

The accompanying sketch (Fig. 53) will enable the student to follow the course of the steam. The slide valve, which in the Plate is shown in the middle of its stroke, and covering both steam ports equally, has moved to the right and uncovered the left hand port, so as to admit steam to the cylinder. In the meantime the hollow internal part of the valve—a part having no connection whatever with the steam surrounding the valve—has opened a passage from the right to the exhaust pipe (shown black), which leads away to the air in a high pressure engine, or to a condenser in a condensing engine. When the piston arrives at the end of its stroke to the right, the valve will have moved to the left, so as to be just ready to open the right-hand port for steam to send the piston back again, and at the same time to open the left-hand side to exhaust.

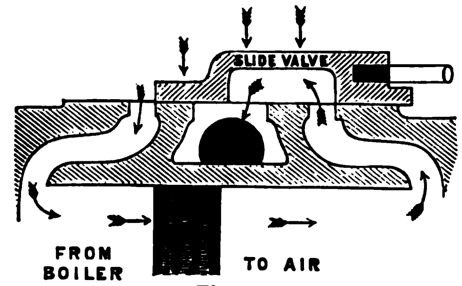


Fig. 53.

The length of stroke of the piston is, in this example, 7 ins., and is the distance travelled by any point in it—say its centre—in moving from one end of the cylinder to the other. Notice that the crank arm (Plate XVIII.) is therefore 3½ in. long between the centre of shaft and centre of pin.

In drawing the cylinder cover it will be well to draw an outline of the piston at the end of its stroke, and then to trace the shape of the cover, so that the piston will clear the cover everywhere by ⅛ in. See "Length of cylinder," page 74.

The whole of the parts are made of cast-iron except the bush of the piston rod. The slide valve itself is sometimes of brass, but more usually of cast-iron.

The motion of the piston inside the cylinder is communicated by the piston rod to the cross-head, and thence through the connecting rod to the crank. The opening in the cylinder end, through which the piston rod passes, is made steam tight by means of a stuffing box, that is, a hollow box for holding packing. The packing, which consists of soft twisted hemp and tallow, or other substitute, is pressed tightly round the rod by means of the gland, or stuffing box cover, which may be screwed up to any required degree of tightness by bolts. A similar arrangement will be noticed for the slide valve rods (Plate XXIV.) and for the pump plunger (Plate XXX.)

A more elaborate form of gland and stuffing box, as fitted for the piston rods of marine engines with inverted cylinders, is illustrated in Fig. 54. The chief feature about this example is the arrangement for tightening up all the nuts of the main gland at once, by merely tightening up one nut. Each nut is cast solid, with a small toothed pinion. The pinions all gear with a toothed ring of brass, which fits round the neck of the gland. By tightening any one of the nuts the toothed ring turns all the other pinions in gear with it in the same direction, and thereby tightens all the nuts at once. This ensures that the gland shall be tightened up evenly, and its axis kept true with the axis of the rod, and thus preventing the possibility of friction between the piston rod and gland, which might otherwise occur.

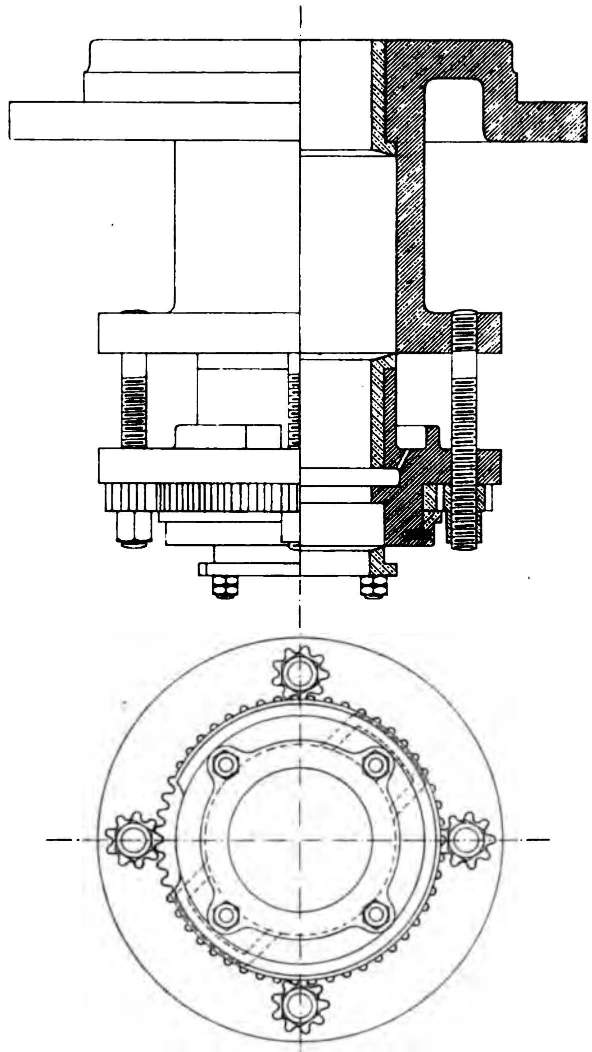


Fig. 54.

To Determine the Proportions of a Steam Engine Cylinder.—Having given the indicated horse-power required, we can obtain the *area of the piston* from the following formula :—

$$\text{Indicated horse-power} = \frac{P\,L\,A\,N}{33000}$$

$$\text{or, area of piston} = \frac{\text{I.H.P.} \times 33000}{P \times L \times N}$$

When P = mean effective pressure of steam in lbs. per sq. in.

L = length of stroke in feet.

A = area of piston in sq. ins.

N = number of strokes per minute.

The *diameter of the cylinder* for a given indicated horse power (I.H.P.) is obtained from the area; thus

$$\text{Diameter} = \sqrt{\frac{\text{area}}{\cdot 7854}}$$

or, the diameter may be found from a table of areas.

Ex. Find the diameter of a cylinder required to develop 80 I.H.P., mean effective pressure 30 lbs.; length of stroke, 2 feet 6 inches ; No. of strokes per min., 200.

$$\text{Area} = \frac{80 \times 33000}{30 \times 2\frac{1}{2} \times 200} = 176 \text{ sq. ins.}$$

$$\text{diameter} = \sqrt{\frac{176}{\cdot 7854}} = 15 \text{ ins.} \quad \textit{Answer.}$$

It will be evident to the student after examining the above formula, that for a given horse-power, so long as the product P × L × A × N remains constant, we may obtain the result by

any number of variations in the values of the terms. Hence the great variety of forms and proportions of engines by different makers to do the same amount of work. In the early engines we had low pressures, massive cylinders, long strokes, and few revolutions per minute. In the modern engine the same power is obtained with high pressures, small cylinders, short strokes, and a high number of revolutions, thus considerably reducing the weight and first cost of the engine.

The first factor P, the mean effective pressure of steam, depends upon the pressure of the steam admitted to the cylinder at the commencement of the stroke, the point of the stroke at which the supply is cut off, and the back pressure against the piston. Its value may be found by drawing an approximate indicator diagram and finding the mean of the ordinates; or by reference to Tables given in works on the Steam Engine; or it may be calculated from the formula:

$$\text{mean effective pressure} = p\,\frac{1 + \text{hyp. log}\,r}{r} - \text{back pressure,}$$

when p = absolute initial pressure of steam; that is, pressure of steam by boiler gauge + 15 lbs.; — loss of pressure between boiler and cylinder.

r = ratio of expansion, = length of stroke in inches ÷ distance travelled by piston in inches before the steam is cut off.

The following Table gives the value of $\frac{1 + \text{hyp. log.}\,r}{r}$

Point of cut off.	Ratio of expansion r	Hyp. log. r	$\frac{1+\text{hyp. log.}\,r}{r}$	Point of cut off.	Ratio of Expansion. r	Hyp. log. r	$\frac{1+\text{hyp. log.}\,r}{r}$
$\frac{1}{10}$	10	2·302	·33	$\frac{6}{10}$	1·66	·507	·907
$\frac{1}{8}$	8	2·079	·385	$\frac{5}{8}$	1·6	·470	·919
$\frac{2}{10}$	5	1·609	·522	$\frac{7}{10}$	1.43	·358	·949
$\frac{1}{4}$	4	1·386	·596	$\frac{3}{4}$	1·33	·285	·966
$\frac{3}{10}$	3·33	1·203	·661	$\frac{8}{10}$	1·25	·223	·978
$\frac{3}{8}$	2·66	·978	·744	$\frac{7}{8}$	1·14	·131	·992
$\frac{4}{10}$	2·5	·916	·766	$\frac{9}{10}$	1·11	·104	·995
$\frac{1}{2}$	2	·693	·846				

Ex. Find the mean effective pressure of steam in the cylinder of a condensing engine when the initial pressure = 30 lbs. and the steam is cut off at one-fourth of the stroke; back pressure = 3 lbs.

By formula—

$$\text{Mean effective pressure} = p\,\frac{1 + \mathit{hyp.\ log.}\,r}{r} - \text{back pressure}$$

$$\text{(by Table)} \quad = 30 \times \cdot596 - 3$$

$$= 14\cdot88 \text{ lbs.}$$

Speed of Piston.—The mean speed per minute depends upon the length of the stroke in feet through which the piston moves, and the number of strokes per minute; or, length of stroke × 2 × number of revolutions. The speed of pistons varies from 200 to 250 ft. per min. for small stationary engines, and from 500 to 750 ft. for marine screw engines. In some instances it exceeds 1,000 ft. per min. for locomotives.

THE LENGTH OF STROKE in ordinary cases varies from $1\frac{1}{2}$ to $2\frac{1}{2}$ times the diameter of the cylinder. In practice many circumstances arise which influence the length of stroke which can be given to an engine, such as the space at disposal, and the character of the work to be done.

It should be noticed that where increased power is obtained by increasing the piston area in preference to increasing the length of stroke, the loads on the working parts and bearings are correspondingly increased, and these parts require, therefore, to be made proportionately stronger.

LENGTH OF CYLINDER inside = length of stroke + depth of piston + twice *clearance* between piston and cylinder cover at end of stroke. The amount of this clearance varies from about $\frac{1}{4}$ in. or less, in small, well fitted engines, to $\frac{3}{4}$ in. in large engines. The clearance must be sufficient to allow for irregularities in the castings, and for wear at the working joints; namely, in the brasses of the cross-head, crank pin, and shaft journals.

THICKNESS OF METAL IN CYLINDERS.—In marine engines for cylinders without liners,

$$\text{thickness} = \frac{\sqrt{D}}{6} + \frac{P\,D}{8000},$$

where D = diameter of cylinder, and P = pressure of steam by boiler gauge.

Thickness of metal in cylinders of compound engines fitted with liners,

$$t = \frac{\sqrt{D}}{7 \text{ to } 8} + \frac{P\,D}{8000},$$

where P = boiler pressure for high pressure cylinder, and P = $\frac{1}{2}$ boiler pressure for low pressure cylinder.

$$\text{Thickness of cast-iron liner} = \frac{\sqrt{D}}{5{\cdot}5};$$

$$\text{Thickness of steel liner} = \frac{\sqrt{D}}{6}.$$

In locomotives the thickness of metal in the cylinder in *sixteenths* of an inch = diameter in ins. × maximum pressure of steam per sq. in. ÷ 120.

CYLINDER COVER BOLTS.—Let d = diameter of bolts, n = No. of bolts, and D = diameter of cylinder, p = maximum pressure of steam.

$$\text{Then, } n\,\frac{\pi}{4}d^2 f = \frac{\pi}{4}\,D^2 p$$

f should not exceed 4000 lbs.

The number of bolts in the cylinder covers is determined more by the necessity for tight joints, especially in high pressure cylinders, than from considerations of strength.

Diameter of bolts for cylinder covers is $\frac{7}{8}$ inch in locomotives, and about 1 inch in marine engines, and the pitch, or distance apart, is about *four* diameters for high pressure, and five to six diameters for low pressure cylinders.

AREA OF STEAM PORTS.—In the ordinary engine the steam port is not only the passage for admission of steam to the cylinder, but also the means of exit for the exhaust steam.

The area of steam ports may be found by the following rule:—

$$\text{Steam port area} = \frac{\text{area of piston} \times \text{speed of piston in feet per min.}}{6000}$$

Having determined the area, the ratio of the length of port to width is determined by making the length from 0·6 to 0·8 the diameter of the cylinder.

Ex. Find the dimensions of the steam port for the cylinder in the above example 15 ins. diameter; length of stroke, 2 ft. 6 ins.; number of strokes, 200.

$$\text{Steam port area} = \frac{15 \times 15 \times \cdot 7854 \times 2 \cdot 5 \times 200}{6000}$$
$$= 14\tfrac{3}{4} \text{ sq. ins.}$$
$$\text{Length of port} = 0 \cdot 7 \text{ diameter of cylinder}$$
$$= 0 \cdot 7 \times 15 = 10 \cdot 5 \text{ ins.}$$
$$\text{Then width} = \frac{\text{area of port}}{\text{length of port}} = \frac{14 \cdot 75}{10 \cdot 5} = 1 \cdot 4 = 1\tfrac{3}{8} \text{ ins.}$$

The whole width of the steam port is opened to allow the steam to exhaust, but only from 0·6 to 0·9 of the port is usually opened for admission of steam to the cylinder.

THE WIDTH OF THE BAR OR BRIDGE of metal between the steam and exhaust ports is not less than 1¼ times the thickness of metal in the cylinder. The danger of a too narrow bridge is an over-travel of the valve, which would open a communication between the boiler and the air through the exhaust passage.

THE WIDTH OF THE EXHAUST PORT is so arranged that when the *valve* is at the extreme end of its stroke, the exhaust port is open an amount equal to the width of the steam port.

THE AREA OF THE STEAM PIPE = ¾ the area of the steam port.

THE AREA OF THE EXHAUST PIPE = the area of the steam port.

Separate slide faces of hard cast-iron or phosphor bronze are sometimes fitted to the cylinder face, and of a thickness equal to about 1¼ times the thickness of metal in the cylinder.

Ex. The stroke of the piston is 7 ins. and its diameter is 7 ins.; find the displacement of the piston per stroke.

Ex. How many cubic feet of steam will be required per hour for the pair of cylinders 7 ins. diameter and 7 ins. stroke, cutting off steam at ¾ of the stroke and making 100 revolutions per minute?

27. PISTONS. PLATES XXIV AND XXXIVA.

In designing pistons we have to bear in mind:—

1. The piston must form a steam-tight division between the two ends of the cylinder.

2. The friction between the piston and cylinder must be as little as possible consistent with steam-tightness.

3. The piston must be as light as possible consistent with strength.

4. The width of the edge of the piston must be sufficient to prevent undue wear of the inner surface of the cylinder.

5. It must be fastened securely to the piston rod.

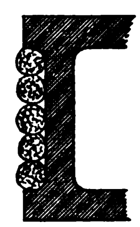


Fig. 55.

In the early days of the steam engine, when steam pressures were very low, pistons were made steam-tight by coiling rope, or *junk*, in a groove on the rim of the piston (Fig. 55); and this method is still adopted for pump buckets, which only require to be water-tight. A common and simple method of packing pistons is that shown in Plate XXIV., consisting of three small steel spring rings of rectangular section, called Ramsbottom's rings. The rings are turned, in the first place, to a diameter a little larger than that of the cylinder they are required to fit; they are then cut in one place, a piece being taken out to enable them to close up to the diameter required,

and when placed in position in the groove of the piston they have a tendency to spring outwards, and thus to render the piston steam-tight.

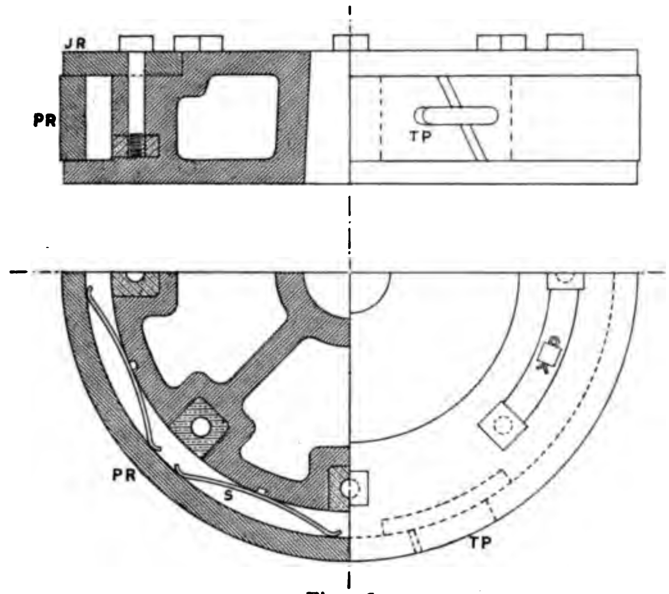


Fig. 56.

PR packing ring. JR junk ring. TP tongue piece. S steel springs.

Another very common method of packing pistons adopted by marine engine makers, illustrated by Fig. 56, consists of one large cast-iron ring PR, which is pressed outwards against the cylinder by means of a series of springs S, placed behind the ring. This ring is cut through by an oblique slit on one side, and tends to spring open as wear takes place. The steam is prevented from leaking through this opening by a brass tongue-piece TP, which is fitted in a groove and placed across the slit. The tongue-piece is secured to a plate fastened to the back of the ring and on one side of the slit.

The ring is held in its place between two flanges, one of which is cast solid with the piston, the other being formed by a loose flat ring called the *junk ring* JR. The junk ring is secured to the piston by screwed bolts, which work into brass nuts inserted in a cavity left for the purpose in the body of the piston.

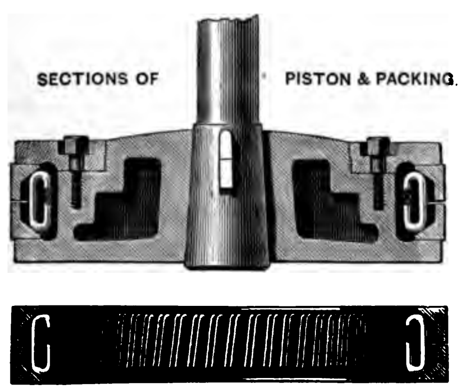


Fig. 57.

The example shown in the Plate XXXIV.A is an example of a high pressure conical steel piston as fitted to a compound marine engine. It will be seen that the piston rod is flanged and secured to the piston by a series of bolts instead of by screwing the piston rod and using one large nut.

Fig. 57 is an illustration of Buckley's Patent Piston. The packing consists of two separate rings with a continuous coiled spring behind it. The action of the coiled spring is to keep the ring steam tight, not only against the cylinder but against the junk ring and flange of the piston.

28. THE SLIDE VALVE. Plate XXV.

Fig. 58 is a sketch of the slide valve shown in the Plate. The valve is in shape somewhat like a hollow inverted dish. Its lower face is made a perfectly true plane surface by the aid of the planing machine, and a surface plate and scraper. This highly finished face slides upon a similarly carefully prepared surface in the steam chest of the cylinder.

When the slide valve is in its middle position it completely covers both steam ports, and prevents admission of steam to either side of the piston. If the length of the valve were exactly the same as the distance between the two extreme edges of the steam ports, then the smallest movement of the valve would open one or other of the ports to steam. The earliest slide valves

were made in this way; but it was found advantageous to increase the length of the valve, so that when in mid position it might overlap the steam ports.

Fig. 58.

In the example given in the Plate the total length of the valve is made $\frac{3}{8}$ in. longer than the distance between the extreme edges of the port, hence it overlaps the port $\frac{3}{16}$ of an inch at each end when the valve is in mid position; in other words, the valve is said to have $\frac{3}{16}$ in. *outside lap*. It also overlaps the inside edge of the steam ports $\frac{1}{16}$ of an inch. It has therefore $\frac{1}{16}$ in. *inside lap*.

In Fig. 59 the lightly shaded valve has no lap. The dark parts show the addition of inside and outside lap by increasing the length of the face of the valve.

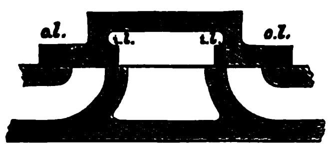


Fig. 59.
o.l. outside lap. *i.l.* inside lap.

The amount by which the valve overlaps the edges of the port when in the middle of its stroke, is called the *lap* of the valve.

The amount of opening of the port for the admission of steam when the piston is at the beginning of its stroke, is called the *lead* of the valve.

It will be noticed that the centre of the valve rod is $\frac{1}{16}$ in. below the centre of the hole in the valve through which the rod passes; this allows for wear on the face of the valve.

To SET A SLIDE VALVE.—Put the crank alternately on its two dead centres. Measure the opening of the port to steam allowed by the valve at each end of the stroke. When these are equal to the lead required in each case the valve is correctly set.

SLIDE VALVE RODS.—The load to be transmitted through the slide valve rod is due chiefly to the friction between the slide valve and the face of the cylinder.

The amount of the load caused by the pressure of steam on the back of the slide valve is estimated as follows:—Length of valve $\times$ breadth of valve $\times$ maximum pressure of steam in lbs. per sq. in. $\times$ the co-efficient of friction, which must be taken at 0·15 to 0·2.

For large engines, diameter of valve rod $= \sqrt{\text{pressure on back of valve}} \div 100$ for wrought-iron, and 110 for steel.

For small engines practical considerations will determine a suitable diameter.

To prevent the slide valve rod from bending or springing, a guide is fitted outside the stuffing box, except in the very smallest engines, consisting of a cast-iron bracket with a bushed hole.

29. ECCENTRICS. PLATE XXVI.

Eccentrics are used when a very small to-and-fro motion is required to be derived from a circular motion of a shaft. They are applied mostly to drive steam engine slide valves, or pumps having a short stroke. The simplest form of eccentric is a circular solid disc, called a sheave, secured to, and revolving with, the shaft, and having its centre "out of centre," or "eccentric," with the centre of the shaft. The sheave is surrounded by a thin metal hoop, or band, called a *strap*, to which the eccentric rod is attached. The circular motion of the sheave is transmitted through the strap and rod, and results in the to-and-fro motion of the valve. This action will be best understood by considering the sheave as a very large crank pin, and the eccentric rod and strap as a simple connecting rod. Then the rectilinear motion of the valve is exactly the same as that which would

be obtained from a small crank the length of whose arm is equal to the eccentricity of the eccentric sheave, that is, the distance between the centre of the shaft and the centre of the sheave. The sheave rotates within the eccentric strap just as the crank pin rotates within the connecting-rod head. The friction of the sheave upon the interior of the strap is, however, excessive, and eccentrics are seldom used where the simple crank can be adopted.

The example of an eccentric given in the plate has an eccentricity of ⅞ in. The dotted circle shown in the drawing, struck from centre C, and passing through E, represents the path of the eccentric centre. Its diameter is 1¾ in., and this distance is equal to the travel of the valve when the valve receives its motion directly from this eccentric without an intermediate link. In order to get the eccentric in its place on the shaft, it has been necessary to make the sheave in halves. The halves are secured together by means of two bolts, which are passed through holes drilled in the sheave, and secured by split cotters. The strap is also made in halves, each half having lugs to take the bolts which secure them together. The upper half of the strap is arranged to receive the foot of the eccentric rod, shown in the right hand bottom corner of the drawing. The rod is secured to the strap by two ½ in. bolts. The lubrication of the sheave is arranged for by casting a small oil cup solid with the strap, and fixing a tube in it through which a cotton wick, dipping into the oil, passes. To show this clearly, part of the strap has been broken away, and the lubricator is shown in section. A small screwed cap with a hole in it forms a cover for the lubricator. To keep the strap in position it is made with flanges overlapping the sheave, the rim of the sheave being grooved to receive them (shown on the sectional view). The strap is made of brass. It is very frequently made of cast-iron, or of cast-iron lined with brass or white metal.

To Find the Position of the Eccentric Relative to the Crank (Fig. 60).—From the centre O, with radius OD = the eccentricity of the eccentric, describe the circle CRD. Draw OC for the position of the crank on its dead centre (that is when the piston is at the beginning of its stroke). From the point O, mark off, on the centre line, CD, the distances O*e*, equal to the lap, and *ef*, equal to the lead of the valve. Then, for rotation in the direction of the arrow, from *f* draw *f*R perpendicular to CD, cutting the circle in R. Join OR. Then R is the required centre of the eccentric, and the line drawn through OR is the centre line of the sheave.

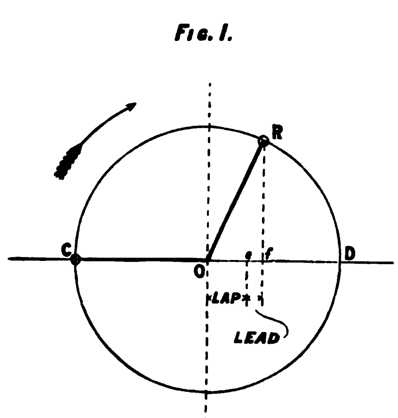


Fig. 60.

The eccentric sheaves might, with advantage, be turned out of a solid collar, forged on the shaft, thus doing away with the separate sheaves, and the necessity of fitting and keying on.

Proportions of Eccentrics.—The function of the eccentric is to move the slide valve to-and-fro on the face of the ports, and in order to do so it must overcome the friction between the valve and the face due to the pressure of steam on the back of the valve.

The resistance to be overcome by the eccentric and its connections is equal to the pressure of steam × area of valve × co-efficient of friction; when co-efficient = 0·15 to 0·2.

The diameter of the sheave is made as small as possible, and it is usually determined by the thickness of metal necessary between the edge of the shaft and the edge of the sheave at the

smallest part, which may be made equal to $\frac{1}{2}$ breadth of sheave + $\frac{1}{4}$ inch. The breadth of the bearing surface of the sheave = $\sqrt{\text{pressure on back of valve}} \div 60$ for locomotives and 40 to 50 for stationary and marine engines.

Ex. The slide valve of a locomotive is 10 ins. long and 15 ins. wide, pressure of steam 150 lbs. per sq. in.; find the breadth of the bearing surface of the eccentric sheave.

$$\text{Breadth of sheave} = \frac{\sqrt{10 \times 15 \times 150}}{60} = \frac{150}{60} = 2\tfrac{1}{2} \text{ ins.}$$

Metal round shaft = $\frac{1}{2}$ breadth of sheave + $\frac{1}{4}$ inch = $1\frac{1}{4} + \frac{1}{4} = 1\frac{1}{2}$
Metal round rim = 0·8 metal round shaft
Breadth of key same as thickness of metal round shaft
Thickness of key = $\frac{1}{2}$ breadth of key.

One or two steel set screws are often used for large eccentrics, to assist the key to resist twisting stresses.

The Eccentric Straps.—Thickness of strap when of brass or cast-iron, about 0·5 breadth; when of wrought-iron or cast-steel, a little less. When a brass liner is interposed its thickness may be $\frac{1}{4}$ thickness of strap.

Diameter of bolts in strap = 0·6 diameter of slide valve rod + $\frac{1}{8}$ in. The thickness of the flanges or lugs of strap are not less than 1·5 thickness of strap. It is well for the sake of both strength and appearance to keep the strap bolts as close up to the sheave as possible. This can be done by increasing the depth of the lugs until the nuts have room to turn.

Diameter of bolts connecting halves of sheave = ·8 diameter of bolts in strap.

Eccentric Rod.—The section of the eccentric rod at the link end is the same as the section of the slide valve rod. The rod then tapers towards the eccentric at the rate of about $\frac{1}{2}$ in. per foot. There are various methods of securing the rod to the strap; a T end to the rod secured by two bolts to the strap is the most common. The size of these bolts is the same as those in the strap flanges. For proportions of fork end of eccentric rods, see link motion.

30. ZEUNER'S SLIDE VALVE DIAGRAM. Plate XXVII.

This diagram (Fig. 2) was first proposed by Dr. Zeuner, of Zurich. It shows very clearly the whole movement of a simple slide valve relatively to the ports during a complete revolution of the crank. It will be best understood by reference to the valve and ports, Fig. 1. First draw the valve and ports, Fig. 1, full size, and then construct the diagram, Fig. 2, below it as follows:—

Draw the centre lines XY, X^1Y^1 at right angles to each other, intersecting in O. From centre O, with radius OR^1 equal to half the travel of valve, describe the circle $R^1R^2R^3$, &c. The dimensions are given in Fig. 1. From centre O, with radius OV = the outside lap, describe the outside lap circle VV^1, &c. From point V, in OR^1, set off VP = the lead. At P raise a perpendicular PP^2, cutting the circle RR^1 in P^2. Join P^2O and produce it to R^7. Then, as before explained, OP^2 is the radius of the eccentric; it is also the position of the centre line of the eccentric when OA is the centre line of the crank. The angle R^3OP^2 is called the ***angular advance*** of the eccentric, being the angle which the centre line of the eccentric has advanced beyond 90° from the centre line of the crank AO. Describe circles on OP^2, OR^7 as diameters; these are called the ***valve circles***. From centre O, with radius OW = the inside lap, describe a circle WW^1. This completes the construction of the diagram.

At the commencement of the stroke the *real* position of the crank arm is AO, and that of the eccentric centre line OP^2; and if these centre lines, with their accompanying valve circles, were made to rotate bodily round the centre O in a direction *opposite* to that of the arrow, it can be shown that the intersection of the valve circles, with the fixed radius OR^1, would give distances OP equal to that which the slide valve has moved from its central position.

A more simple method, giving the same result, is to rotate the radius OR^1 in the direction of the arrow, while everything else remains fixed. Then, if OR^1 is the assumed position of the crank at the commencement of the stroke, OP, the part of OR^1 intercepted by the valve circle, is the distance which the valve has moved from its central position; and this distance includes OV the lap, and VP the port opening, shown shaded, which is in fact the lead of the valve. The opening of the port to exhaust at the same instant is given by the length DW^3. Following the path of the crank round the circle in the direction of the arrow, if a radius be drawn through B, the intersection of the valve and outside lap circles, we have OR the position of the crank when the opening of the port to steam *begins*. The valve has then travelled a distance OB, equal to the lap, from its central position. When the crank reaches the dead point, as before explained, the port is open a distance VP. Continuing the rotation of the crank, the valve continues to open the port wider, until at the position OP^2 of the crank the port is open to the full extent allowed by the travel of the valve. (It will be noticed that the distance V^2P^2, which measures $1\frac{3}{8}$ in. on the diagram, is less than the width of the steam port, which is $1\frac{5}{8}$ in.; thus showing that the port is not fully opened for *admission* of steam, though, as will be shown below, it is in due time fully opened to exhaust). On continuing the rotation the port now begins to close, until at OR^4, drawn through the intersection of the circles at V^4, the valve closes the port. OR^4 is therefore the position of the crank at the point of "cut off." The steam is now no longer admitted to the cylinder, but presses the piston forward by its own expansive force until the crank reaches OR^6, drawn through the intersection W^1 of the inside lap and valve circles, when it commences to escape into the air, or the condenser. R^5 drawn tangent to the valve circles, or at right angles to OP^2, is the position of the crank when the valve is in its middle position. The port opens wider and wider to the exhaust till the crank reaches OM, when the port is fully open. At OR^7 the valve has moved a distance R^7Q beyond the edge of the port, thus leaving the port wide open to exhaust till the crank reaches ON, when the port begins to close. The distance WQ = width of port = $1\frac{5}{8}$ in. At OR^8 the line drawn through the intersection W^2 of the inside lap and valve circles, the exhaust port closes, and any remaining steam is compressed until the crank reaches OR, when steam is again admitted, and the same cycle of operations repeated.

The position of the piston for a given length of connecting rod may be easily determined by drawing a circle about centre O, to represent the crank pin path, to some convenient scale, say 3 ins. = 1 ft. Produce the lines OR^1, OR^2, &c., to meet this circle, and from these points on the circumference as centres, with a radius equal to the connecting rod to the same scale, cut the centre line XY produced by arcs. These will give the positions of the piston for any required position of the crank pin. From centre O mark off a distance, on OY produced, equal to the length of the connecting rod to the same scale. This will represent the centre of the stroke.

Suppose we have given the travel of the valve, the point of cut off and the lead of the valve, and we require to find the lap of the valve.

Draw the circle radius OA = half the travel of valve. Draw the line OR^4 for the given position of the crank at the point of cut off, assuming the crank to start from OR^1. From centre R', and with radius $R'S$ = the lead, draw an arc, as shown in Fig. 2. Through R^4, draw a line R^4R *touching* this arc and cutting the circle in R. Through O, draw OP^2 at right angles to R^4R. This line bisects the angle R^4OR. From centre O, draw a circle touching the line R^4R. OV^2 is the lap required; and the angle AOP^2 is the angle between the centre lines of crank and eccentric to obtain the required cut off. Also VP = RS, the *lead* required.

Ex. A steam port is 10 inches long and 1⅝ ins. wide, the travel of the valve is 5 ins., and the lap of the valve 1⅛ ins.; find the maximum area of port opening to steam. *Ans.* 13·75 sq. ins.

31. THE LINK MOTION. PLATES XXVIII. AND XXIX.

The object of the Link Motion is to provide a means for reversing the engine. In simple engines, which always run one way, the single eccentric is sufficient; but for reversing, something further is necessary. Referring to the notes on the Eccentric, it was shown that if the circle C R Q, Fig. 61, be the path of the eccentric centre, then R, obtained as before explained, is the position of the centre of the eccentric sheave for the position O C of the crank when the engine rotates in the direction of the arrow. For rotation in the *opposite direction*, Q is the corresponding position for the eccentric centre. Hence, in order to reverse an engine with *one* eccentric, we should require to be able to remove the centre of the eccentric from R to Q. In any case, we must have some arrangement which is equivalent to this. The object is accomplished in the Link Motion by having two eccentrics secured to the shaft side by side, one having its centre at R, and the other its centre at Q; and we have now to connect these with the valve rod in such a way as to enable us to bring the valve under the influence of either eccentric as may be required. By referring to the Plate XXVIII., or to Fig. 62, it will be seen that the upper part of the rods of the respective eccentrics are attached to a slotted bar called the link L. Within the slot is a little solid block which is attached to the slide valve rod SVR by a pin. By a simple arrangement of levers, a forward or backward sliding motion of the link connects the valve with the backward or forward eccentric as required. The slide valve rod, which is connected directly with the slide valve, can only move up and down vertically, and the extent and nature of this movement depends entirely upon the movement communicated to them through the eccentric rods and link.

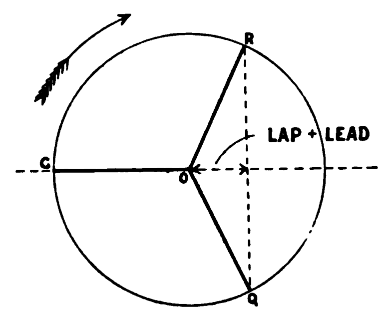


Fig. 61.

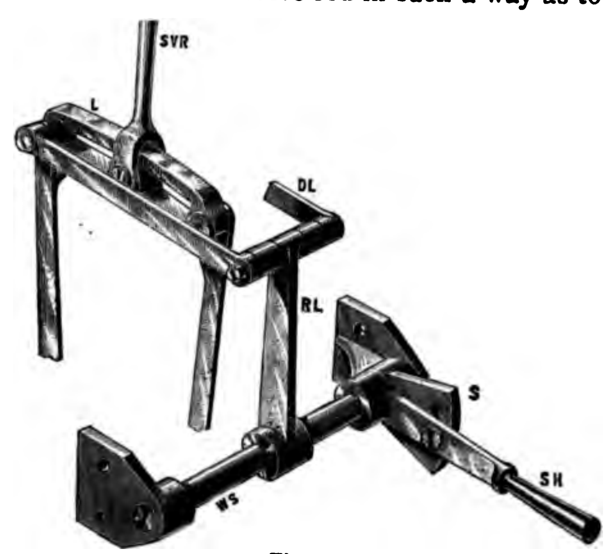


Fig. 62.
SVR slide valve rod. L link. DL drag link. RL reversing lever
WS weigh shaft. SH starting handle. S sector.

To go ahead, the handle SH, Fig. 62, is pushed down, and the link, therefore, pulled forward. This brings the slide-valve under the influence of the eccentric, whose centre is R, Fig. 61, and the motion of the valve is nearly the same as though it were connected directly with this eccentric. To reverse the engine, the handle SH is pulled up, the link thrown backward, and the slide-valve put under the influence of the eccentric, whose centre is Q. When the link occupies the position midway between its two extreme positions, the valve is influenced equally by both eccentrics, with the result that the engine will not run in either direction.

The nearer the link is to its mid-position, the less is the travel of the valve, and the earlier the steam is cut off from the cylinder. This enables the engineer to regulate with great nicety the consumption of steam and the speed of his engine.

To Draw the Link Motion (scale half full size), Plate XXVIII.—First draw vertical centre line A B, and upon it, towards bottom of paper, take a point for centre of crank shaft. From this point draw the dotted circle with $\frac{7}{8}$ in. radius, to represent path of eccentric centres. Through the centre draw a horizontal line C D, and measure from it, upon the dotted circle, for centres 1 and 2, distances equal to $\frac{5}{32}$ in. (See full-size drawing for these centres in right-hand bottom corner of plate.) As there are two cylinders and two slide-valves, there are two link motions and four eccentrics for the engine. (See General Drawing.) It is sufficient to draw one link motion with its two eccentrics, which are struck from centres 1 and 2. From centre 1, with radius equal 2 ft. 1 in., draw an arc through F upon which to place centre of pin at top of rod. Repeat a similar curve through G from centre 2. Take a point on each curve $4\frac{1}{4}$ ins. on each side of centre line A B. These two points are the centres of the pins which connect the eccentric rods to the link. From either of these pin centres, F or G, with radius equal to 2 ft. 1 in., cut the centre line A B towards the crank shaft. This point (just above E) is the centre from which the link is struck. Complete the link by arcs from this centre. The dimensions of the eccentrics and eccentric rods are obtained from Plate XXVI.

The eccentric sheave rotates with the shaft, and quite independently of the strap. (See Eccentric.) Referring to the front eccentric, the sheave is drawn about the centre line joining the centre of the shaft E with its own centre 1. The strap is drawn by first joining centre 1 with point F in the link for the centre line of the eccentric rod. Then through centre 1 draw line *m n* at right angles to centre line of eccentric rod F 1. This line *m n* is the centre line of the strap. The other eccentric is drawn similarly. In the side view of the eccentrics one view is projected from the backward eccentric. The sectional view is not projected, but is placed in the best position to show the true form of the section.

Link Motion Diagrams, Plate XXIX.—Dimensions of Parts:—Eccentricity of eccentrics, $1\frac{3}{4}$ ins. Length of eccentric rod from centre of sheave to centre of pin, 20 ins. Length of link, $8\frac{1}{2}$ ins. Distance of centre A (Fig. 2) from near end of link, $1\frac{3}{4}$ ins. Length of drag link (radius C*m*), $11\frac{3}{8}$ ins.

This drawing is given to enable the student to follow the path of the points of importance about a link motion. The dimensions chosen are somewhat different from those given in the Launch Engine details. The eccentricity has been increased, and the eccentric rods have been shortened, so as to give a somewhat exaggerated result compared with those occurring in actual practice.

Draw the diagrams for mid-gear as in Fig. 1, for full forward gear as in Fig. 2, and for full backward gear by a similar method to that used for the forward gear.

The process is as follows :—The inner circle represents the path of the eccentric centre, and the outer circle the path of the crank pin. First, with the crank on the lower dead centre, mark the positions 1F and 1B ½ inch above the centre line, these being the positions of the forward and backward eccentrics respectively, relatively to the crank.

As before explained the eccentric, which influences the admission of steam to the cylinder, always leads or is ahead of the crank by some angle greater than 90°. When the crank rotates in the direction of the arrow, F is the leading eccentric; when the crank rotates in the opposite direction, B is the leading eccentric.

To obtain a succession of positions of the link during one revolution of the crank shaft when the link is in mid gear :—Divide the crank pin path into any number of equal parts, say twelve, beginning from the lower dead centre. These are successive positions of the crank pin during a single rotation. To find the corresponding positions of the forward eccentric F, divide its circular path into the same number of equal parts (12), commencing at 1F, and going round the circle, as shown at 2F, 3F, &c. Similarly for the backward eccentric, commence at 1B, and divide the circle into twelve equal parts, 2B, 3B, &c.

Now find the position C of the reversing lever C D by the dimensions given, and from C as centre, with radius of drag link (11⅜ ins.) describe the arc *m n*. The method of obtaining a suitable position for C is described further on. It will be seen, by reference to Fig. 62, that the end of the link to which the drag-link is attached must move in an arc about the centre C, Fig. 1 in the Plate. From centres 1F, 2F, 3F, &c., with radius equal to the length of the eccentric rod (1 ft. 8 in.) measured from centre of *sheave* to centre of pin at extremity of rod, cut the arc *m n* in a succession of points, and number them 1, 2, 3, &c., according to the number of the centre from which they are struck. From centres 1B, 2B, 3B, &c., with the same radius, draw short arcs on the opposite side of the centre line A E, and number them as before. Now make a templet from a piece of thin wood or cardboard, Fig. 63, with arc $a\,c\,b =$ arc of the link struck with radius of 1 ft. 8 in.. $a\,b = 8\frac{1}{2}$ ins., and $c =$ centre of link; lay the templet with *a* on forward rod position No. 1, on arc *m n*, and *b* on opposite or backward rod position No. 1. Draw the arc *a c b*, and repeat this for each successive position. Join Nos. 1, 2, 3, &c., of the backward rod pin, and we see that its path is something like a figure 8. Remembering that the slide valve rod receives its straight line motion from these various movements of the link, we can here see (Fig. 1 on the Plate) what the extent of that motion is when the link is in mid-gear, namely, *s t*. It will also be noticed that in mid-gear the extreme positions of the *valve* occur for the positions 1 1 and 7 7 of the *link*, and these coincide with the dead centres 1 and 7 of the *crank*, or the points at which the *piston* is at the respective ends of its stroke. The point A, mid-way between *s* and *t*, is the centre of the pin in the link block. If from centre A, with radius A L = the lap of the valve, a circle be drawn as shown, the distance from *s* or *t* to the edge of the lap circle, measured on the centre line, shows the extreme port opening when the link is in mid-gear. From this it will be seen that even when the link motion is in mid-gear there is still some opening of the port to steam. Referring to Fig. 2, the difference in the port opening in the two cases can be compared. The curved figure between *s* and *t* is obtained by marking the middle point of the link *c* (Fig. 63), for each position of the link, and joining these points by a curve. This indicates the

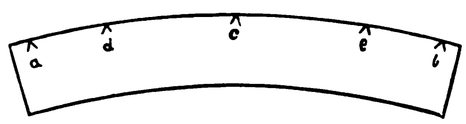


Fig. 63.

amount of "slotting" or longitudinal slipping motion of the link upon the block. It will be noticed that the corresponding figure $s't'$ in Fig 2. is very narrow, showing that the rubbing motion of the link upon the block is very small in forward gear, the most common working position of the link; in fact, the nearer the link block is to the point of suspension, the less the slotting motion. When equal efficiency of the link is desired in both backward and forward motion, the link is suspended from the centre by securing the pins for suspending the link to plates bolted on the sides; but when the slotting motion of the link is to be reduced to a minimum in full forward gear, the link is suspended from the end next the forward eccentric rod.

SUSPENSION OF THE LINK.—The slotting motion above referred to is due to the compulsory movement of one end of the link on the arc described by the drag link, together with the oscillating movement of the link caused by the combined motion of the two eccentrics. The length of the drag link and the position of the centre C from which the arc is struck, should evidently be such as to reduce the objectionable slotting motion to a minimum.

To find the centre of the point of suspension:—Draw the successive positions of the link in full forward gear, and again in full backward gear, without the restraints of the drag link, so that there shall be no slotting motion. To do this for full forward gear keep the point d of the templet (Fig. 63) on the centre line through the valve rod, and mark the successive positions of the point of suspension a of the link. This will give the unrestrained path of a for full forward gear; repeat this for full backward gear, by keeping point e of the templet (Fig. 63) on the centre line, and again tracing the path of a, the point of suspension. Now, with a radius equal to the assumed length of the drag link, find a centre such that the arc described from it will, as nearly as possible, *bisect the irregular curved figure* for full forward gear. This centre should be chosen for the full forward centre C′ (Fig. 2) of the reversing lever. Similarly find the centre for the arc in backward gear. From each of these centres, with radius equal to the length of the reversing arm, describe arcs intersecting in a point which will give the centre of the weigh shaft. Usually, for practical reasons, in fixing the weigh shaft centre a compromise is made in favour of the forward gear position, as some degree of slotting motion is not generally so serious a defect in backward as in forward motion.

The diagram Fig. 3 is an application of the Zeuner valve diagram to the Link Motion. The arc P P^1 P^2 Q represents the link in miniature, and P^1 P^2, &c., are positions which the link may occupy relatively to the block for various grades of expansion. P^3, in the middle of the arc, represents the link in mid gear.

To draw the figure:—From centre S, radius S R = the lap of the valve, draw the lap circle R L; and with radius S Y = the eccentricity of the eccentric, draw the circle X M N Y.

Draw S P and S Q so that the angles M S P, N S Q = the angular advance of the eccentrics. The curve P P^1 P^2 Q, which should be parabolic, may be drawn with a

$$\text{radius} = \frac{\text{P Q} \times \text{length of eccentric rod}}{2 \times \text{length of link}}$$

where P Q = distance between centres of eccentric sheaves (as from 1 F to 1 B, Plate XXIX.), and lengths of rod and link are measured between centres, as on a skeleton drawing.

This is a useful approximation, first proposed by Mr. M'Farlane Gray.

On S P describe a circle. Then if the slide valve rod were connected *directly* with the end of the eccentric rod, without the intervention of a link, the motion of the two would be identical, and

the circle on SP, as explained in the Zeuner valve diagram, would give the port opening, &c., for any position of the crank. But on comparing Fig. 3 with Fig. 2, it will be seen in Fig. 2 that when the link is in full forward gear, the link block, which is connected with the slide valve rod, is $1\frac{3}{4}$ ins. from the eccentric rod, and hence the travel of the valve will be less than that of the eccentric rod pin. In Fig. 3, if the length ($P\,P^1$) be made to bear the same ratio to P Q as $1\frac{3}{4}$ ins. bears to $8\frac{1}{2}$ ins., and a circle be described on $S\,P^1$ we may, by drawing lines through the intersection of this circle with the lap circle, obtain the point of admission, lead of valve, port opening at any point in the crank pin path, point of cut off, &c., when the link is in full forward gear. In other words, the valve will move as if connected directly to an eccentric whose eccentricity is $S\,P^1$, and angular advance $M\,S\,P^1$. The same particulars may be found when the block is at any intermediate position (P^2) by drawing the circle $S\,P^2$, or, when the link is in mid-gear, by drawing the circle $S\,P^3$.

PROPORTIONS OF THE LINK MOTION.—The distance between the centres of the eccentric rod pins in the link ends should not be less than three times the throw of the eccentric.

Breadth of link = diameter of slide rod.
Thickness of bars = diameter of eccentric rod pins.
Diameter of eccentric rod pins = $\frac{3}{4}$ diameter of valve rod.
Diameter of block pins = $1\frac{1}{8}$ diameter of valve rod.
Length of block = $1\frac{3}{4}$ diameter of block pin.

POSITION OF THE STARTING GEAR.—Looking along the centre line of the crank shaft and towards the head of the vessel, the cranks usually rotate in the direction of the hands of a watch when the vessel is going ahead, with a right-handed propeller; when they rotate in the opposite direction, for going ahead they require a left-handed propeller.

When the link is suspended from its centre the starting gear may be on either side, according to convenience; but when the link is suspended from one end, in order to keep the drag link as long as possible, the weigh shaft is placed on the side opposite to the forward eccentrics. With the starting gear as shown in the coloured plate, the engine will require a left-handed propeller. By removing the starting gear to the other side of the engine, and attaching the drag link to the opposite end of the link, the engines will rotate in the opposite direction, and a right-handed propeller may be used.

"Twin screw" engines, which are much used in war vessels, consist of two separate propeller shafts, driven by separate engines, rotating in opposite directions, with right and left-handed screws; the screws being made to revolve outward from the tops during ahead motion.

32. PUMPS. PLATE XXXI.

The plate shows exterior and sectional views of a pair of pumps, both of which are the same size in every particular.

One is used for pumping water into the boiler, and is called the Feed Pump; the other, when required, pumps water out of the bottom or bilge of the launch, and is called the Bilge Pump. The whole of the parts are made of gun-metal.

The two plungers are solid, and are connected together by a slotted cross-head.

Motion is given to the plungers by a pin which is attached to a disc on the end of the engine crank-shaft (see coloured General Drawing). The pin passes through a hole in the block B. While the pin on which the block rides describes a circular path, the block itself slides up and down in the vertical groove of the slotted cross-head, and at the same time pushes the cross-head bodily from side to side horizontally. In this way the circular motion of the pin is transformed into the reciprocating motion of the plungers.

The pump is supplied with water by a pipe which is bolted to the suction end of the pump, and which leads below the surface of the source of water supply.

THE ACTION OF THE PUMP may be explained as follows:—Referring to the sectional view, suppose the plunger to be at the extreme right of its stroke, and the whole interior to be full of air. When the plunger moves to the left, the pressure on the suction valve being lessened, the air in the supply pipe lifts the valve and flows into the barrel. The pressure of the air in the pipe is now less than before, and accordingly the pressure of the atmosphere on the exterior surface of the water forces water up the suction pipe to such a height as to make the pressure inside the pipe balance the atmospheric pressure outside. When the plunger returns, the suction valve is closed, and air is forced out through the delivery valve. Each time the stroke of the plunger is repeated, the water will rise in the supply pipe, until at last it reaches and fills the barrel. Now when the plunger returns, it forces water instead of air through the delivery valve.

The height of the column of water which will balance the pressure of the atmosphere is 34 feet; that is, a column whose weight is equivalent to 15 lbs. per square inch. In practice, however, it is impossible to obtain a perfect vacuum, hence the supply should never be drawn through a greater height than 25 feet. The height to which, or the pressure against which, the water can be delivered from the pump is only limited by the strength of the material of the pump.

THE VALVES are conical disc valves, and they are guided in rising and falling in their seatings by four thin webs. The delivery valve is prevented from rising too far out of its seating by a stop on the cover. A projection on the bottom of the delivery valve forms the stop against which the suction valve lifts.

The studs and nuts for tightening up the glands are placed the reverse way, to be clear of the reciprocating cross-head.

The *lift* of a valve regulates the amount of feed which can pass through the opening, and it can be shown that when the lift equals one-fourth the diameter of the valve, the area of the opening to escape round the edge of the valve is equal to the area of the circular aperture in the seating.

Let the valve have a *flat* seating as shown in the Fig. 64. Then area of opening to flow of water round edge of valve $= \pi \times d \times l$; and area of opening in valve seating $= d^2 \times \frac{\pi}{4}$. Then when these two values are equal we have—

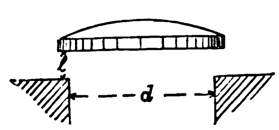


Fig. 64.

$$\pi \times d \times l = d^2 \times \frac{\pi}{4};$$

$$\text{or, } l = \frac{d}{4}.$$

That is, when the area of the openings are equal, the lift $= \frac{1}{4}$ the diameter.

When the valve and seating are coned to 45°, as is usually the case, the opening for escape of water round edge of valve is less than the lift. In Fig. 64A the perpendicular $a\,c$ is less than the lift $c\,b$, for the angles at c and b being 45°, $a\,c : b\,c : 1 : \sqrt{2}$; or, width of opening is to lift of valve as $1 : 1{\cdot}4$.

It is only usual to give so great a lift as $\frac{1}{4}\,d$ to small valves; with large valves the lift is usually much less.

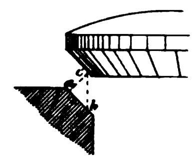


Fig. 64A.

The nett quantity of water to be pumped into the boiler is the amount required to make up for what has been used as steam in the engine, together with losses through priming, leakage, blowing off, &c. To determine the quantity of water used as steam in a cylinder, multiply the area of the piston in feet, by the speed of the piston in feet per minute, and by 60; the result gives the number of cubic feet of steam used per hour. Divide by the specific volume of steam at the given terminal pressure, and the result gives the corresponding volume of water. It is usual, however, in order to provide for emergencies, to make the pump capable of throwing from 2 to $2\frac{1}{2}$ times, or in large marine engines 3 or 4 times, this amount.

Pumps always deliver less water than corresponds to the piston displacement and number of strokes; and the ratio between the quantity of water really thrown, and the volume described by the plunger, gives the *efficiency* of the pump.

The suction pipe of pumps may be provided with a cock or valve to regulate the supply of water to the pump. A similar fitting may be applied to the delivery pipe of pumps, but when so fitted there must also be a feed escape valve to prevent an accident should the valve on the delivery pipe be closed.

AIR VESSELS or Chambers are fitted to pumps close to and beyond the delivery valve. The air in the water collects in this vessel, and forms a cushion or spring, which enables the water to be delivered in a continuous stream, and which greatly improves the action of the pump. The cubic contents of the air vessel is about 5 times the piston displacement per stroke for short delivery pipes, to 10 times for long delivery pipes.

Area of waterway through valve seating = diameter^2 × ·7854 − area of webs.

Capacity of pump in cub. ins. per minute = area of end of plunger × length of stroke, in ins. × number of strokes per minute.

Average efficiency of good pumps = capacity × 0·8.

1 cub. ft. of fresh water = 62·5 lbs., or 1000 ounces.

1 cub. ft. of sea water = 64 lbs.

1 lb. of water = ·016 cub. ft.

1 gallon of water = 10 lbs., or 0·16 cub. ft.

Questions.—1. What is the length of the stroke of the plunger in Plate XXXI.?

2. Calculate the volume displaced by the feed pump plunger at each stroke.

3. What should be the maximum lift of the suction and delivery valves?

4. Suppose the efficiency of the feed pump to be 0·8, find the cubic feet of fresh water thrown by the pump per hour, the number of revolutions of the pin in the slotted crosshead being 60 per minute.

33. EXERCISE IN SHADING. PLATE XXXI.

For instructions, see page 9.

34. GENERAL DRAWING OF THE STEAM LAUNCH ENGINES.
PLATES XXXII. AND XXXIII.

It is assumed that the student has prepared a complete set of the engine details. These should be ready at hand for reference, as the whole of the details are now to be built up as one complete machine. The drawing should be made on a double elephant sheet of paper, and three views of the engine shown, namely, the two given in the Plates, and a *plan* placed underneath the side elevation. It is expected that the student's knowledge of drawing will, at this stage, enable him to construct the plan without difficulty. A convenient scale for the drawing is 3 ins. = 1 foot.

First set out the principal centre lines, the most important of which are as follows:—

Distance between centres of cylinders	1 ft. $2\frac{3}{4}$ ins.
Distance between centre of cylinders and centre of crank shaft ...	2 ft. $10\frac{3}{4}$ ins.
Distance between centre of piston and centre of cross-head pin ...	1 ft. $4\frac{3}{4}$ ins.
Distance between centre of crank shaft and centre of cross-head pin when cross-head is in centre of stroke = length of connecting-rod, namely:—	1 ft. 6 ins.

Begin by drawing the cylinder, shown in section, and place the piston and cross-head for this sectional view in the middle of the stroke. The outline of the other cylinder may now be drawn, the small portion of the cross-head visible being $3\frac{1}{2}$ ins. higher (that is, half the stroke) than the one already shown. The lagging or covering of the cylinder is composed of non-conducting material, covered with strips of mahogany $1\frac{1}{2}$ ins. wide. The lines representing these strips are obtained by first setting them off in position on the plan (when it is drawn), and projecting them in the elevation. Now, about the centre line of the crank shaft the four bearings may be drawn ready for placing the crank shaft in position. The correct position of the cranks cannot be obtained until the side elevation of the engine is drawn, and this view should now be proceeded with, beginning at the top and working downwards, and carefully projecting the parts from one view to the other. If it is not intended to colour a General Drawing, it is usual to show as much detail as possible upon it, by dotting in the parts which are hidden from view. Even when it is intended to colour the drawing, it will be found necessary to dot in at first many of the parts which are hidden, in order to project them in their proper positions on other views.

Draw (and where necessary dot) in position on the end elevation the cranks, connecting rods, and eccentrics, and both of the link motions, in order to obtain the correct projections of these parts on the side elevation. It should be observed that the crank pins are not on the vertical or horizontal centre lines, but are obtained, as described in the article on connecting rods, by putting the cross-head in the centre of the stroke. Then the distance from the centre of the gudgeon to the centre of the crank shaft is equal to the length of the connecting rod. Draw a circle from the centre of the crank shaft with radius equal to the radius of the crank, namely, $3\frac{1}{2}$ ins. This represents the path of the crank pin. From the centre of the gudgeon, when in its mid position, with radius equal to the length of connecting rod, draw an arc passing through the centre of the crank shaft, and intersecting the crank pin path. This intersection gives the correct position of the crank pin when the piston is in the middle of its stroke. The other crank makes 90° with

QUADRUPLE EXPANSION MARINE ENGINES.

MESSRS. FLEMING AND FERGUSON, ENGINEERS, PAISLEY.

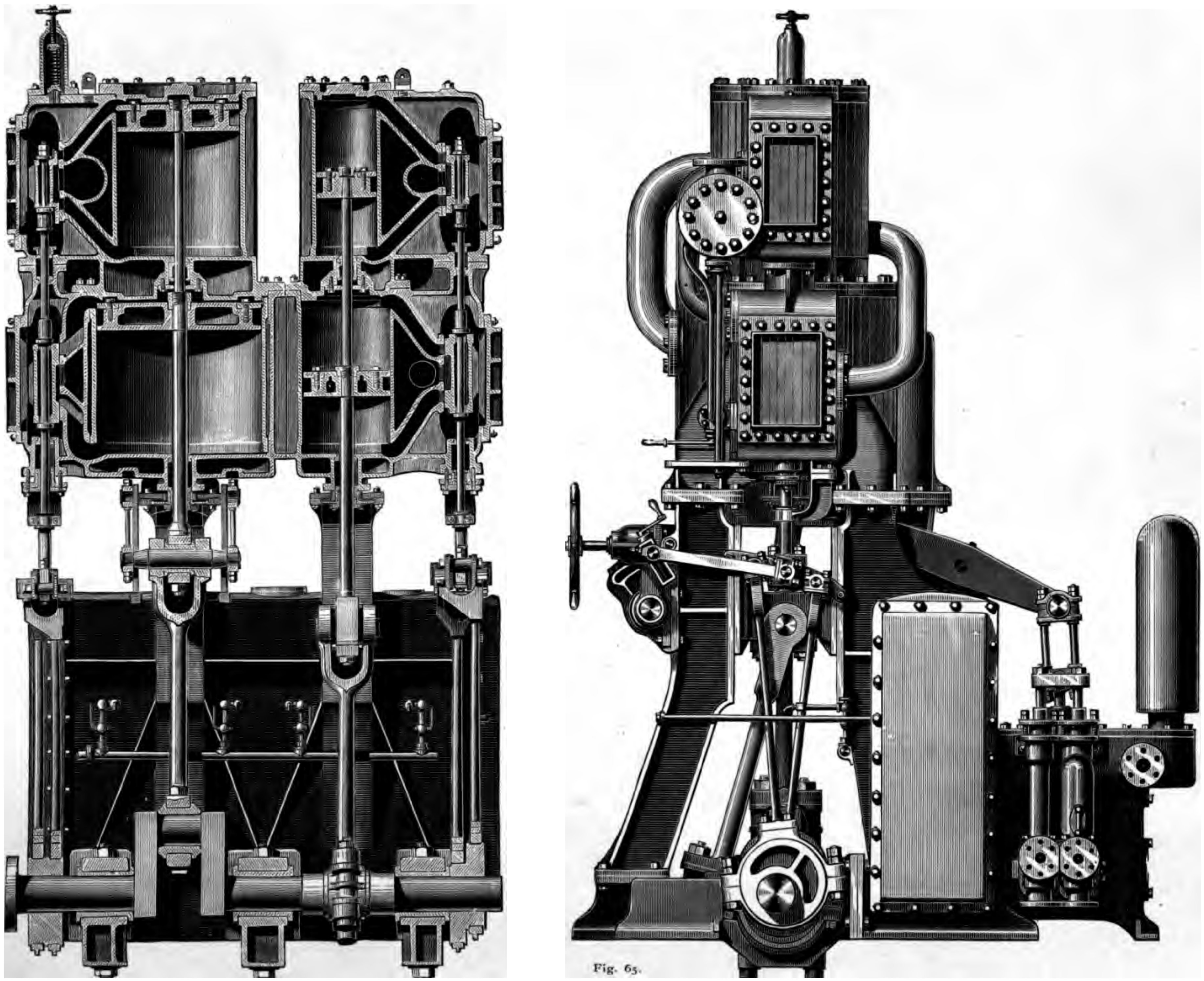


Fig. 65.

the first. Frequently the cranks are drawn, one on the horizontal centre line, and the other on the vertical. In that case one piston is somewhat *beyond its mid position* towards the crank, and the other is at the extremity of its stroke.

For remarks on the position of the starting levers see Link Motion (page 85).

To Colour the General Drawing of the Steam Launch Engines.—Commence by shading all the wrought-iron, cast-iron, and steel parts with a ground tint of Indian ink, by the rules and methods already described (p. 9). The ground tint for brass may be burnt umber, instead of Indian ink, as the latter gives the brass an objectionable green tint when finished. Burnt umber, however, cannot be put on nicely by a series of washes, but it should be laid on at once, to the depth required, graduation being obtained by lines of colour varying in depth of tone. The colour used for the mahogany lagging of the cylinder is burnt sienna, and it is treated in the same way as the burnt umber. The graining of the mahogany is done with a deeper tint of the same colour. Having now completed the shading of all the parts, the special colour for each metal is now laid as a thin even wash, care being taken to leave white lines of light where necessary.

35. QUADRUPLE EXPANSION MARINE ENGINES.

Fig. 65.

The example given in Plates XXXII. and XXXIII. is that of a pair of engines with high pressure cylinders, each of which receives steam direct from the boiler, and exhausts it into the air. This is the simplest possible form of engine, and, as such, it has been chosen for our exercises in drawing.

Though high pressure cylinders only are still largely used for locomotives and stationary engines, they are by no means an economical arrangement, as they fail to take full advantage of the economy to be derived from working the steam expansively.

The extent to which the expansion of steam has been carried, within the past few years, is attributable to the increased working pressure of the steam, which it has been found possible to obtain by improved boiler construction.

A few years ago no marine boiler was made to work at more than 30 lbs. pressure to the square inch, and then the expansion of the steam took place in one cylinder only, exhausting from this directly into the condenser. But, as steam pressures and temperatures were increased, it was found necessary, in order to prevent great variations of temperature in one cylinder, to distribute the expansion over two cylinders. Thus one cylinder received its steam from the boiler, and exhausted it into a second larger cylinder, where it might do further work before being exhausted into the condenser. This arrangement of one high and one low pressure cylinder forms what is called the "compound engine."

But as pressures and temperatures have continued to increase, it has been found more economical and convenient, for the same reason as that stated above, namely, to avoid great variations of temperature in one cylinder, to use three, and even four, cylinders, called respectively triple and quadruple expansion engines.

An engraving of the quadruple expansion engines of the steam yacht "Grace Darling," which appeared in the *Engineer* of March 23rd, 1888, is here added, by the kind permission of the makers, Messrs. Fleming and Ferguson, engineers, Paisley, and the editor and proprietors of the *Engineer*, to illustrate the extent to which high-pressure steam may be economically expanded.

Two views of the engine are given. The first shows a section through all four cylinders, showing the slide valves, steam ports, and the construction of the pistons and stuffing boxes. The steam enters the smallest or high-pressure cylinder only, direct from the boiler; and it is then successively expanded to the second, third, and fourth cylinders (their order being that of their diameters) by means of the pipes shown on the other view; and finally, into the condenser, shown as a rectangular box, forming part of the engine framing.

The diameters of the cylinders are 10½ ins., 14 ins., 20 ins., and 28 ins. respectively; the stroke is 20 ins., and the indicated horse-power 360. The whole of the low-pressure cylinder, and the bottom of the third cylinder, are jacketed. The upper cylinders form the covers for the lower. The lower cylinders have hand holes in front, to allow of their pistons being sighted, and the junk ring pins felt without disturbing the upper cylinders. The packing between the cylinders is of the self-adjusting spring metallic type, and will last for years without attention. The crank shaft is 5½ ins. diameter, with cranks at right angles; it is forged from the best wrought-iron scrap, and has three bearings, 10 ins. in length. The condenser has 390 sq. ft. of cooling surface.

Air pump	12 ins. diameter, 12 ins. stroke.
Circulating pump	7 ins. diameter, 12 ins. stroke.
Feed pump...	2¾ ins. diameter, 12 ins. stroke.
Bilge pump	2¾ ins. diameter, 12 ins. stroke.

The heating surface in the boiler is 752 sq. ft., and the grate surface 27 sq. ft.
Working pressure of steam in boiler, 200 lbs. per sq. in.

36. EXPANSION VALVE GEARING. PLATE XXXIV.

The advantage of working steam expansively—that is, of cutting off the supply at some point before the end of the stroke of the piston—was recognised at an early period in the history of the steam engine. As has been already explained, a certain degree of expansion may be obtained with the common slide valve, by the addition of lap, and the greater the lap the earlier the cut off. But with a single slide valve, connected directly to the eccentric, the steam cannot be cut off before the piston has travelled about two-thirds of the stroke. The amount of additional lap necessary to cut off at any earlier point requires so much additional angular advance of the eccentric, as to result in a too early opening and closing of the exhaust; thus, on the one hand wasting steam, and on the other increasing compression. When any greater degree of expansion is desired, some form of Link motion, or a separate expansion valve is used. Hitherto the Link motion has been most generally in favour with locomotive and marine engineers, both for reversing and for varying the degrees of expansion; but for stationary engines, especially when not requiring to be reversed, a separate expansion valve is used; and the example given in the Plate illustrates a very common and efficient arrangement, known as Meyer's Expansion Valve Gear. It consists mainly of two plates, sliding on the back of the main valve, and whose distance apart may be varied by means of a right- and left-handed screw.

We will first consider the action of the valves by reference to a simple figure (Fig. 66). MV is the main or distribution valve, which is an ordinary valve, with the addition of pieces at the ends, which provide steam passages (*pp*) through the valve.

Consider the expansion valve EV moving to the left, then, when *e* reaches *n*, cut off takes place, and the port in the main valve MV is closed, till on the return stroke *e* again passes *n*. Evidently by increasing the distance between the two plates of the expansion valve, or, in other words, by lengthening the valve, the distance *s* between *e* and *n* is decreased, and cut off takes place earlier.

The lengthening or shortening of the valve to cut off at any required point in the stroke can be regulated by means of the screwed valve spindle.

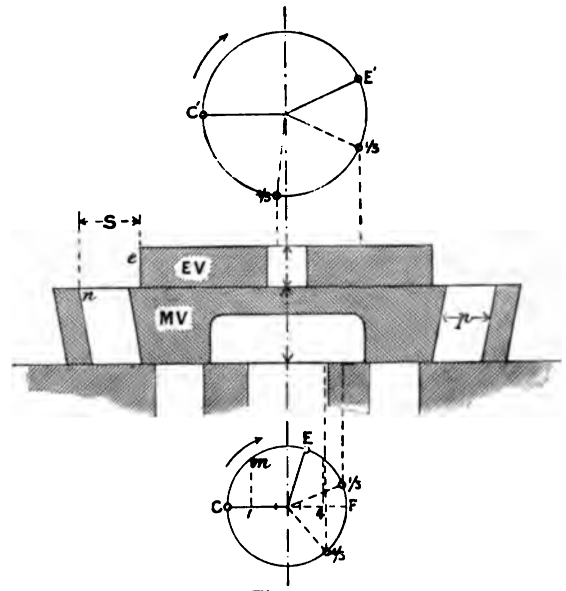


Fig. 66.
MV main valve. EV expansion valve.

The main valve acts as an ordinary simple slide valve, and the engine might be worked with it alone, if the expansion valve were entirely removed. The main valve, however, is provided with only a small amount of lap, and would therefore, by itself, supply steam to the cylinder through the greater part of the stroke. The existence of the expansion valve does not in any way affect the action of the exhaust through the main valve.

The relative motion of the two given valves may be conveniently found by the following practical method:—

1. Draw the steam and exhaust ports for the cylinder face to the dimensions given on the Plate (XXXIV.) on a sheet of paper.

2. Make a templet of the main valve of thin board, or stiff paper, and mark on it distinctly its centre line. (Make the sliding edges perfectly straight and parallel).

3. Take a strip of board or stiff paper, to represent the expansion valve, and mark on it a centre line only.

4. Draw the circle C E F below the cylinder ports representing the path of the centre of the main valve eccentric sheave (diameter $2\frac{1}{2}$ ins.), and mark on this circle the position of the eccentric centre E (angular advance = 16°) when the crank is on its dead centre; also its positions when the crank is at the required limiting points of cut off, say, at positions $\frac{1}{5}$th to $\frac{4}{5}$ths of the stroke of the piston.

To do this divide the diameter C F of the circle C E F into five part, and raise a perpendicular from Division 1 to meet the circumference at *m*. This is the position of the crank pin, approximately, when the piston has moved through $\frac{1}{5}$th of the stroke. Measure the arc C *m*, and set the eccentric forward from E by the same arc to the point marked 1/5. Raise a perpendicular from this point to the cylinder face. The centre line of the main valve will coincide with this line when the piston is at $\frac{1}{5}$th of the stroke. Proceed similarly to obtain the centre of the eccentric and valve at $\frac{4}{5}$ths of stroke.

5. Above the expansion valve (Fig. 66) describe a circle C′ E′ ($3\frac{3}{8}$ ins. diameter) representing the path of the centre of the expansion valve eccentric sheave, and draw the centre E′ of the

eccentric (angular advance 65°) when the crank C′ is on the dead centre. Find the position of this eccentric centre at $\frac{1}{5}$th and $\frac{4}{5}$ths of the stroke by setting it forward by the same angles as were found for the main valve eccentric.

Let fall perpendiculars from these positions, as shown.

Place the main valve templet so that its centre line coincides with the perpendicular from the $\frac{1}{5}$th position on the lower circle. It will be seen that the main valve does not then close the steam port. Now place the expansion valve so that its centre line coincides with the $\frac{1}{5}$th position from the upper circle, and draw a line to represent the edge *e* of the expansion valve, so that it will close the port of the main valve for admission of steam. This line gives the extreme length of the expansion valve to cut off at $\frac{1}{5}$th.

To find its length to cut off at $\frac{4}{5}$ths, place the centre line of each valve templet against the perpendicular from the $\frac{4}{5}$ths position of its respective valve circle. Draw a line to represent the edge *e* of the expansion valve, so as to just cover the port in the main valve. This gives the length of the valve to cut off at this point of the stroke. By this method, having given the main valve, it is plain how to design an expansion valve to cut off at any point of the stroke.

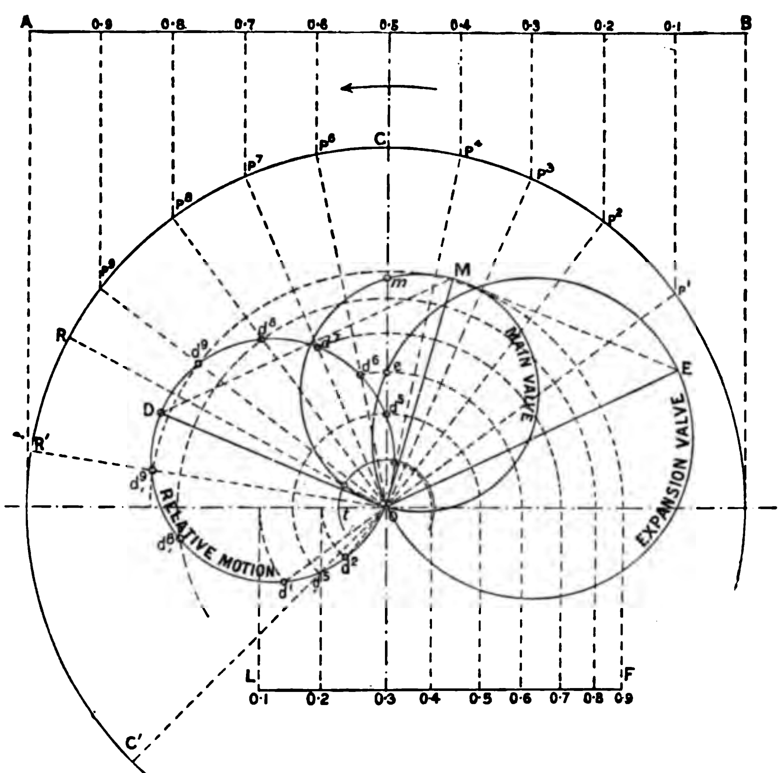


Fig. 67.

It must be particularly noted that the plates composing the expansion valve must be sufficiently wide to prevent *re-opening* of the port in the main valve before the closing of the cylinder port. To test this point, adjust the expansion valve for its highest degree of cut off, that is, when

the expansion plates are most widely separated, then place the main valve so that it is just re-closing the cylinder port, and the expansion valve in its corresponding position relative to it; then the after-edge of the expansion valve plates should at least completely cover the main valve port.

All the movements of the valves in Meyer's Expansion Valve Gear can be very easily determined by the application of the Zeuner valve diagram, without the aid of models.

For a detailed and exhaustive description of such diagrams, the student is referred to Dr. Zeuner's "Valve Gears," translated by Prof. Klein.

The following is the application of the diagram to the example given in the Plate.

The setting of the eccentrics relative to the crank is shown in the Plate XXXIV., and the valves are represented in their middle position, disconnected from their eccentrics.

In the diagram, Fig. 67, O E = $1\frac{1}{8}$ inch is the eccentricity of the expansion valve eccentric, full size, with an angular advance of 65°, and O M = $1\frac{1}{4}$ inch is the eccentricity of the main valve eccentric, with an angular advance of 16°, and if circles are described upon these lines as for a simple valve, the motion of the valves on each side of their centres are given for any position of the crank, as explained at page 79. Thus, let any position of the crank O C intersect the circles O M and O E at m and e. Then O m is the distance of the main valve from the centre of its stroke, and O e the distance of the expansion valve from the centre of its stroke. Hence $e\,m$ is the distance between the centres of the two valves. Set off O $d^5 = e\,m$, and repeating this for every position of the crank, we obtain a circle d^1, d^2, O, d^5, d^6, &c., coinciding with the circle O D.

To find the diameter and position of the circle O D :—Join M E, and from M draw M D parallel to E O, and O D parallel to E M. Then O D M E is a parallelogram, and O D is the diameter and position of the circle which is called the *relative motion* circle; and the distance cut off by it as O d^5 from any crank position O C, is the distance between the centre of the expansion valve and the centre of the main valve for that position of the crank.

Now, we saw, from Fig. 66, that cut off and re-admission take place when e reaches n; in other words, when the centres of the valves are at a distance apart equal to s. Therefore, if a circle be struck from centre O, with radius s = O d^5, cut off will take place when the crank is at O C, because O C is the crank position, drawn through d, the intersection of the relative motion circle by the arc with radius s, and re-admission will take place when the crank has moved forward to C¹ through O d_1^5. But in an efficient cut off gear the value of s must be variable and capable of adjustment, each different value giving a different degree of cut off. The effect of such alteration is very clearly seen by the aid of this diagram.

To find the value of s (Fig. 66) to cut off at any given point in the stroke. Draw the circle O C (Fig. 67) with any convenient radius, outside the valve circle, to represent the crank pin path, and draw A B = 2 O C to represent the path of the piston. Divide A B into any required number of equal parts, say 10; these are the piston positions. To find the corresponding crank positions, drop perpendiculars on the crank pin path and draw radii O P^1, O P^2, &c. Suppose it is required to find the value of s to cut off at 0·8, or $\frac{4}{5}$ths of the stroke. The crank O P^8 intersects the relative motion circle at d^8; then O d^8 is the value of s to cut off at $\frac{4}{5}$ths of the stroke. The port in the main valve is closed, while the arc d^8, d_1^8 is described, and at d_1^8 the port in the main valve is re-opened, and steam would be re-admitted to the cylinder, were it not for the fact that the main valve has closed the cylinder port at O R, where the crank passes through the intersection of the circle O M, and the lap circle O l.

When $s = O\, d^9$ the port in the main valve is closed, and steam is cut off at O P^9, but the port in the main valve is re-opened at O R^1 for admission of steam. This, however, is permissible, since the main valve closed the cylinder port at O R. Care must be taken that O R^1 shall always fall behind O R, otherwise steam would be admitted to the cylinder *twice* in one stroke.

If the value of s be taken greater than O D, the expansion valve becomes useless, merely contracting the port in the main valve, instead of closing it. To cut off at O P^8 the value of s is zero; that is, the expansion valve just closes the port in the main valve when both valves are in mid position. If it is desired to cut off at a still earlier point of the stroke as at 0·2, the value of s becomes negative, that is, the expansion valve overlaps the port in the main valve when both valves are in mid position. In the Fig. 67 the crank position O P^2 on being produced intersects the relative motion circle at d^2. Then O d^2 is the distance by which the expansion valve plates overlap the main valve ports, when the valves are in their middle positions, to cut off at 0·2, or ⅕th of the stroke.

The values of s for a cut off at the several points of the stroke are all brought down on the line L F in Fig. 67, full size, and these determine the movements which must be given to the two plates by the screwed spindle to obtain the required space s for a given point of cut off. Meyer's valve gearing may be regulated as desired while the engine is in motion, and the arrangements for doing this are carried by the bracket B, which is supported by arms from the end of the steam chest. Here the expansion valve spindle, after passing through the stuffing box (not shown), is square, and works to and fro through a square hole in the bush of the bracket B. When the wheel W, at the extreme left, is turned, the bush and screwed valve spindle turn with it, the effect of which is to extend or bring together the expansion valve plates; while, in the meantime, the to-and-fro motion of the spindle is not in any way interfered with. Part of the bush is screwed on its outer surface, as shown, and this thread gears with a piece of a nut which works in a groove, and carries a small pointer, or index, whose position on a graduated scale attached to the outside of the bracket shows the point of cut-off for which the expansion valve is set. The graduated scale shown on the Plate XXXIV. is obtained from L F. (Fig. 67.) The joint in the spindle, shown to the right of the expansion valve, enables the valve spindle to turn on its axis without interfering with its connection with the eccentric rod.

37. THE SCREW PROPELLER. Plate XXXV.

The Screw Propeller may be regarded as a short piece of a screwed bolt cut out of a bar of enormous diameter, the threads being cut excessively thin and excessively deep. (See Fig. 69B.) The threads form the blades, and as they are of great pitch, and the screw is short in the direction of the axis, the blade appears as a narrow strip of the deep thread. A single thread will give one blade, a double thread two blades, and so on.

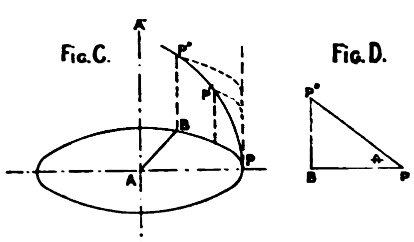


Fig. 68.

Geometrical Principles.—(*a*) Let the vertical straight line A A^1, Fig. 68c, intersect a horizontal plane in A, and let the point P revolve about the fixed point A in the plane, and at a uniform distance from it; then the path of P is a *circle*.

(*b*) Let the point P move parallel to A A^1; then the path of P is a *straight line*.

(*c*) Let now these two motions be combined, both being uniform; then the point P will describe a curve P P′ P″, which is called a ***helical,*** or ***screw*** curve.

(*d*) The length of the forward movement of the point P in making one complete revolution, measured parallel to the axis, is called the ***pitch*** of the helix.

(*e*) If the radius of revolution of the point P be uniform, then the helix described will be on the surface of a cylinder.

(*f*) If the radius of revolution of the point P be variable, the helix described by the point will be on some other surface; as, for example, that of the cone or sphere.

(*g*) If the motion of the point P in the direction parallel to the axis, and also its rate of rotation be uniform, the pitch of the helix is uniform; but if its rate of motion in the direction parallel to the axis *increase,* while its rate of rotation remains uniform, the pitch is said to be *axially expanding.*

(*h*) If the pitch of the helix increase as the radius of revolution of the point increases, the pitch is said to be ***radially expanding.***

We have hitherto been considering the generation of helical curves by points; we will now consider the generation of helical *surfaces* by lines.

(*i*) Let the line A P, Fig. 69B, be a horizontal line, perpendicular to A A′, and let it revolve about the line A A′ at A; then A P will generate a ***horizontal plane.***

(*j*) Again, let the line A P move along A A′ without any rotation about A A′; then the line A P will generate a ***vertical plane.***

(*k*) Let now both these movements be combined, both being uniform; then the surface swept out or generated by the line A P will be a ***helical, twisted, or screw surface;*** and this surface is similar to the working surface of a propeller blade of uniform pitch (Fig. 69), or of a square-threaded screw, or to the plastered under surface of a winding staircase.

(*l*) The distance travelled along the axis A A′ by the generating line A P, in making one complete revolution, is called the ***pitch*** of the thread.

Figs. A and B illustrate the application of the above principles to the drawing of a screw propeller.

To students who have drawn the examples of screws (Plate VII.), this diagram will require no explanation.

The outer edges of the blade ***a′ b′*** and ***c′ d′***, Fig. A, are shown at ***a b*** and ***c d***, Fig. B.

At ***e f*** we have another view of the same two blades, turned through 90°.

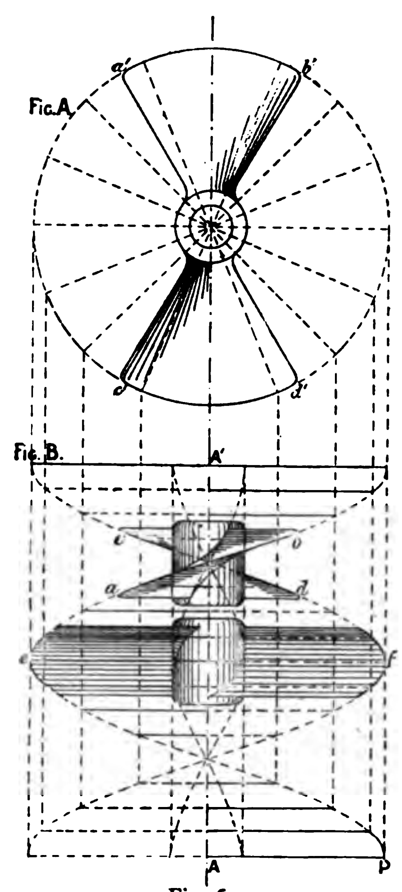


Fig. 69.

ANGLE OF THE SCREW.—Let the line B P, Fig. 68D, be equal in length to the arc B P, Fig. 68C, and let B P″ be equal in both figures. Join P″ P; then the angle θ at P, namely, P″ P B, is the angle of the screw at radius A P. If B P (Fig. 68D) be made equal to the circumference of the blade at a given point, and B P″ equal to the pitch, then the angle θ (P″ P B) is the angle of the blade at

that point. See also Figs. 17 and 18, page 35. It will be evident from the figure that the smaller the circumference BP, the greater the angle at P becomes, provided the pitch B P″ remains constant. Thus the angle made by the edge *a b* of the blade (Fig. 69B) with a plane at right angles to the axis, increases as the blade approaches the centre or boss.

The angle θ of the blade at any point is obtained from the formula :—

$$\text{Tan } \theta = \frac{\text{pitch}}{\text{circumference}} = \frac{\text{pitch} \div 6{\cdot}28}{\text{radius}}$$

The LENGTH OF THE SCREW is the extreme length of the blade measured *in the direction of the axis* of the propeller shaft.

The form of propeller blades has varied very greatly, and continues to vary, with different makers from time to time. On this point Mr. Sennett (in his work on the Marine Steam Engine) says: "The effect of the form of blade has not yet been fully ascertained, but it would appear that the shape suitable for one ship does not always prove equally efficient for another ship, and in our present state of knowledge it would be difficult to predict with certainty the form of blade that will be most suitable for any particular ship. As a rule it would appear that form has no peculiar value as regards propulsive efficiency, though it may have some influence on the amount of vibration produced."

The example chosen as an exercise for the student is a right-handed, two bladed propeller, 7 ft. diameter outside, and 7 ft. uniform pitch.

One face only of the propeller blade is made of the true geometrical form, namely, the after face of the blade, which is its acting surface. The form of the forward face or back of the blade depends upon the thickness of metal at various parts of the blade necessary to provide sufficient strength.

First draw the centre line A′ B′ (Plate XXXV.) representing the axis of the propeller shaft, and then draw to the dimensions given (Fig. 1), which is the form of blade decided on in the present case when the blade is extended or flattened out. Divide the blade into any convenient number of equal parts, as at *a b c d e f*, and draw horizontal lines through these points. From the point *o* on the centre line of the shaft A B, set off *o p* equal to the pitch, divided by $2 \times 3{\cdot}1416 = \frac{7 \times 12}{6{\cdot}2832} = 13\frac{3}{8}$ ins. This gives the same result as if *o f* were made equal to the circumference of the circle through *f*, and *o p* equal to the pitch. (See also Fig. 68 D.) From the point *p* draw lines through *a b c d e f*; these lines represent the angles of the blade at the various sections at distances *o a*, *o b*, &c., from the axis. From the points *a b c*, &c., set off on the lines *p a p b*, &c., the length of the sections made by the horizontal line through *a b c*, &c., on the extended blade, as shown by the arcs.

To obtain the thickness of the sections we commence the other view (Fig. 2). Here the thin vertical sectional strip is supposed to represent a section through the centre line *o a* of the extended or flattened blade, for on no other assumption could we obtain a plane section giving thickness of metal normal to the surface. The left-hand line *o′a′* of this vertical section (coinciding with the centre line of the figure, Fig. 2) is the generating line of the acting surface, and it will be seen that, in the present example, this line is not straight, but is bent over at the end towards the forward end of the ship.

The thickness of blade at tip is $\frac{3}{8}$ ins., and at the lowest section $2\frac{1}{2}$ ins.; and the section Fig. 2 is drawn as shown, the dimensions giving the required taper and curve of blade. Then continuing the horizontal lines through *a b c*, &c., on Fig. 1 to Fig. 2, we obtain the thickness of

metal at thes everal sections of the length of the blade. Transfer these thicknesses to *a b c*, &c., on Fig. 1, and draw arcs through the points for the back of blade, leaving an appreciable thickness of metal at the edge of the blade.

Before finishing Fig. 2, it will be convenient to proceed next with Fig. 5. The point o'' in Fig. 5 would be the plan of the whole line $o'a'$, Fig. 2, if it were straight and vertical; at present it is the plan of that part of it from the bottom of the blade to section *c*.

Draw, therefore, through o'', Fig. 5, lines $c''c''$, $f''f''$, &c., parallel to the lines *p c*, *p d*, *p e*, *p f*, Fig. 1, and set off on these lines the breadths of the blade at these sections, as given by the dimension Fig. 1. Thus $f''o''f'' = 18$ ins. The point a' the tip of the blade, Fig. 2, is bent forward $1\frac{3}{8}$ in. From o'', Fig. 5, with radius $1\frac{3}{8}$ in., draw an arc; then draw a line parallel to *p a*, touching this arc, to represent the top of the blade, and set off the distance $a''a''$ equally on each side of the perpendicular to $a''a''$ through o''. The plan of the blade may now be completed by drawing the curve through the points $a''b''$, &c., as shown. To complete the blade on Fig. 2, drop perpendiculars from each of the points $a''b''$, &c., Fig. 5, on the line AB, as shown at $f''f^2$; then $o''f^2$ is the breadth of the projection of the blade, set off on each side of f', Fig. 2. The position of the points at every other section on Fig. 2 is obtained similarly. The breadths of the elevation of the blade at the horizontal sections of Fig. 3 are also obtained from Fig. 5; thus drop perpendiculars on to the vertical centre line, as $f''f^3$, then $o''f^3$ is the breadth of the elevation of the blade on one side of the centre line, at the horizontal section f'''. The other points of this elevation are obtained in a similar way. The boss and remaining portions of the diagram are drawn from the dimensions.

Chapter VII.

BOILER WORK.

38. AUTOMATIC STOP VALVE. PLATE XXXVI.

Where there is a series of boilers all communicating with one main steam pipe, the stop valve of each boiler may be made self-closing, so that if from any cause the pressure in one boiler should fall below that in the others, the valve will close and disconnect the boiler from the rest. When in position, these valves are placed on the front of the boiler, with the spindle *horizontal,* so that the weight of the valve itself has no tendency to close it. As shown in the Plate, the valve is closed, the screwed bush which is attached to the wheel being screwed down upon the shoulder on the spindle, thus entirely preventing the valve from lifting. By turning the wheel, and thus lifting the bush from the shoulder, any excess of pressure on one or other side of the valve will cause it to open or close automatically. A small handle on the end of the spindle provides a means of moving the valve by hand. The main steam pipe, which is of copper, is made in lengths extending from stop valve to stop valve. Each length is secured by a flange at one end to one stop valve, and at the other end enters the next stop valve through a gland and stuffing box, an arrangement which permits of expansion and contraction of the pipe. The whole of the parts of the valve are made of gun-metal.

Exercise. Design, after the same pattern, a *common stop valve,* namely, one which opens and shuts by hand only, for a steam pipe $2\frac{1}{2}$ ins. internal diameter. Make the spindle the same diameter throughout (namely $\frac{7}{8}$ inch), and screw the top of the spindle instead of having a screwed bush, which may be dispensed with. A screwed hole at the top of the bracket will form the nut for the spindle. No handle will be necessary above the wheel.

39. SAFETY VALVES. PLATE XXXVII.

The safety valve provides for the safety of boilers, by allowing the steam to escape when its pressure reaches a certain limit. The safety valve is kept in its place on its seating by means of heavy weights, or by a weight at the end of a long lever; or, as in marine engines and locomotives, by means of a strong spring.

The following is a description of the example given in the Plate:—The steam passes into the valve box at *a*, and presses against the under side of the valves *b b*. When the pressure of the steam on the under surface of the valves is sufficient to overcome the springs, which are regulated to lift at a certain pressure, the valves lift, and the steam escapes through *c* into the air. The valves are lifted from their seats by hand, when necessary, through the bar G, by the four levers K^1, which bear against the collars M. The collars are solid with the valve spindles S. The spindles are attached to the valves by a split pin, and they terminate at the top in a square end, so that by means of a handle, the valves can be turned round on their seats without being lifted.

The hollow compression nut N is screwed down, and compresses the spring to the amount necessary for the required pressure. The ring or metal stop P is then adjusted, so that the nut tightens upon it. The tube within the spring, ending in a flange at the bottom, keeps the spring in position.

The springs are of steel; all the rest is of gun-metal. The area of the valve, according to the Board of Trade rule, should be not less than ½ sq. in. per sq. ft. of fire grate surface.

The size of the steel of which springs are made is found by the following rule:—

$$d = \sqrt[3]{\frac{W \times D}{C}}$$

where d = diameter, or side of square, of spring steel in inches.
W = load on spring in lbs.
D = diameter of the spring from centre to centre of wire.
C = 8,000 for round steel, 11,000 for square steel.

From this formula we may obtain the value of any other term of the equation.

Ex. Find the equivalent dead weight W of a spring constructed of ¾-inch diameter steel, mean diameter of coil 4 ins.

$$W = \frac{C\,d^3}{D}$$

$$= \frac{8{,}000 \times \cdot 75 \times \cdot 75 \times \cdot 75}{4}$$

$$= 843 \cdot 75 \text{ lbs.}$$

It will be observed that the strength of springs is in the ratio of the *cubes* of the diameter of the wire.

To find the compression of the spring necessary to produce a given pressure on the valve, the following formula is used:—

$$\text{Compression} = \frac{W \times d^3 \times n}{s^4 \times E}$$

where W = total pressure required on valve.
d = mean diameter of spring.
n = number of coils.
s = diameter or thickness of steel in *sixteenths* of an inch.
E = 30 for square steel, and 22·8 for round.

THE LEVER SAFETY VALVE.—To find the weight W, or length of lever A B, for a given pressure of steam.

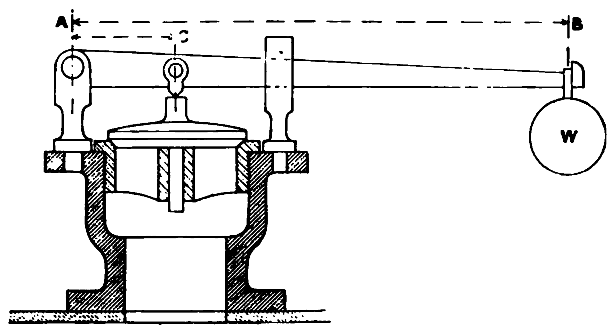


Fig. 70.

Let A B = length of lever from fulcrum A to centre of weight W.

A C = distance between centre of valve and fulcrum.

W = weight at end of lever.

w = weight of lever, acting at centre of gravity of lever, assumed at centre of lever.

P = pressure of steam per sq. in.

a = area of valve.

V = weight of valve.

(i.) If the effect of the weights of valve and lever be omitted, we have, when valve is just about to lift,—

Downward pressures = upward pressures.

$$\underset{\text{Pressure on valve due to W.}}{W \times \frac{AB}{AC}} = \underset{\text{Upward steam pressure.}}{Pa}$$

$$W = Pa \times \frac{AC}{AB}$$

(ii.) Taking the effect of weights of valve and lever into account (which should always be done where accuracy is required) we have, when valve is about to lift,—

Downward pressures = upward pressures.

$$\underset{\text{Pressure on valve due to W.}}{W \times \frac{AB}{AC}} + \underset{\text{Pressure due to weight of lever.}}{w \times \frac{AB}{2AC}} + \underset{\text{Weight of valve.}}{V} = \underset{\text{Total upward pressure on valve.}}{Pa}$$

Ex. Let it be required to find weight W at end of lever when A B = 36 ins., A C = $4\frac{1}{2}$ ins., w = 5 lbs., V = 2 lbs., P = 100 lbs. and a = 5 sq. ins.

From (i.) omitting weight of valve and lever we have

$$W = (100 \times 5) \times 4\tfrac{1}{2} \div 36$$
$$W = 62\tfrac{1}{2} \text{ lbs.}$$

From (ii.) including weight of valve and lever

$$\left(W \times \frac{36}{4\frac{1}{2}}\right) + \left(5 \times \frac{36}{2 \times 4\frac{1}{2}}\right) + 2 = 100 \times 5$$
$$8W + 20 + 2 = 500$$
$$W = 59\tfrac{3}{4} \text{ lbs.}$$

These results show that if a weight of $62\frac{1}{2}$ lbs. was placed on the lever, instead of the proper weight $59\frac{3}{4}$ lbs., the valve would not blow off at 100 lbs. as required.

A suitable length of lever A B for a given weight W may be obtained from the same equations.

At least two valves are fitted to each boiler. The valve seating may be either flat as in the Plate (XXXVII.), or coned to 45° as in Plate IV. A bearing surface on the edge of the seating, $\frac{1}{8}$ in. in width, is found to be quite sufficient, and to answer better than a wider surface.

The length of furnace should not exceed 7 feet, otherwise it becomes difficult to stoke. It will be noticed that the back of the combustion chamber slopes a little inwards towards the top. This enables the steam to rise more freely.

The space allowed between the tubes is 1 in., and the tubes are arranged in vertical rows to allow of the boiler being properly cleaned internally.

STEAM ROOM.—It is important to have as large a reservoir of steam as possible above the level of the water in the boiler, to prevent too great fluctuations of pressure. The water level should be at least 7 ins. above the top row of tubes.

To find the cubic contents of the steam space:—Find the area of the segment of the circle occupied by the steam, and multiply by the internal length of the boiler; and from this substract the contents of the stays which occupy part of the steam space.

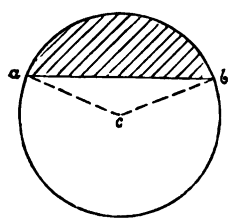


Fig. 71.

To find the area of the segment of a circle, Fig. 71:—Area of whole circle $\times \frac{\text{angle } a\,c\,b}{360}$ — area of triangle *a b c*.

To give the front and back plates of shell the necessary stiffness, large circular plate washers, 10 ins. diameter, are riveted on to outside of plates.

The maximum stress allowed on these stays is 8,000 lbs. per sq. in. for stays under $1\frac{1}{2}$ ins. diameter, and 9,000 lbs. for stays over $1\frac{1}{2}$ ins.

For further information on boiler stays see page 43.

Manholes are placed on the top and front of the boiler, to get at the upper and lower parts of the furnaces for cleaning and repairing. The furnace bars are of wrought-iron, and in three lengths, sloping towards the bridge $\frac{3}{4}$ in. per foot. Distance between bars $\frac{1}{2}$ in., maintained by widened ends of bars.

PROPORTIONS FOR MARINE BOILERS.—Steam required by compound, triple, or quadruple expansion engines equals area of piston of low pressure cylinder in square feet $\times$ speed of piston in feet per minute $\times 60 =$ cubic feet of steam per hour. Divide this result by the specific volume of steam at the given terminal pressure, that is, its volume compared with the volume of the water from which it is produced, and the result will give the cubic feet of *water* required. Add to this say 25 per cent. for waste and contingencies.

The *volume* of water in cub. ft. $\times$ 62·5 = *weight* of water in lbs.

The *weight* of water $\div$ 9 = coal consumed per hour in lbs.

Coal consumed per hour $\div$ 20 = square feet of grate area; or, grate area = 0·12 sq. ft. per I.H.P.

Heating surface of furnaces and combustion chambers = 24 to 29 sq. ft. per sq. ft. of grate.

Heating surface of tubes = 20 to 24 sq. ft. per sq. ft. of grate.

40. THE MARINE BOILER. Plate XXXVIII.

The function of the boiler is to provide the means of generating steam from water by the application of heat. Its essential elements are:—Internal water and steam space; a furnace, and means of providing as far as possible for the complete combustion of the fuel; efficient means of transferring the heat produced to the water; a chimney, to cause a draught and carry away the products of combustion; fittings, to carry the steam from the boiler to the engine, to provide the boiler with fresh supplies of water, to show the height of water in the boiler, to indicate the internal pressure, to relieve the boiler of excess of pressure; means of getting inside the boiler, to clean and examine it; a blow-off, or means of emptying the boiler.

Description of the Plate. The boiler is of the cylindrical, multitubular type, fired from one end, with three furnaces. The products of combustion in the furnaces are carried forward by the draught into the combustion chambers C C, and thence through the tubes in the direction of the arrow to the front of the boiler, whence they pass up the funnel.

The Outside Shell is 12 feet $1\frac{3}{8}$ inch extreme diameter, and 9 feet $5\frac{13}{16}$ inch extreme length. The plates are of steel, $\frac{13}{16}$ inch thick, in three rings united together circumferentially by double-riveted lap joints. The longitudinal seams are treble-riveted. The end plates are made in three pieces, and are joined together by double-riveted lap joints, and flanged to meet the shell and the furnace flues.

The Furnaces are 3 feet inside diameter, constructed of Fox's corrugated steel plates $\frac{1}{2}$ inch thick. They are flanged at the back end and riveted to the combustion chambers.

The Combustion Chambers are flat on the top and are supported by wrought-iron girder stays, details of which are given on Plate X. The back and sides of these chambers are stayed with $1\frac{3}{8}$-in. screwed stays, shown enlarged on the Plate, fitted with nuts on both ends.

The boiler contains 200 tubes, 3 ins. diameter outside, of which 42 are stay tubes. The stay tubes are of wrought-iron, $\frac{5}{16}$ in. thick, and screwed into the plates, shown in detail on the Plate. The remainder of the tubes are of brass.

Longitudinal stays, $1\frac{7}{8}$ in. diameter, steel, pass through the steam space from end to end, and support the front and back plates of shell.

Fire Grate area = area of grate × number of furnaces = $(3 \times 6) \times 3 = 54$ sq. ft.

The heating surface.—The effective heating surface of a marine boiler is obtained by finding the sum of the following areas:—

1. Area of furnace above level of fire bars.
2. Area of sides and crown of combustion chamber above level of bridge.
3. Area of back tube plate, less area of holes for tubes.
4. Area of surface of tubes, namely, the area obtained by multiplying the external circumference by the length *between* the tube plates. The area of the front tube plate is omitted.

The area of the contracted part of the furnace at the bridge is made $\frac{1}{4}$th of the grate area. In the combustion chamber the gases part with some of their heat, and their volume is thereby reduced; hence area through tubes may equal $\frac{1}{6}$th of grate area, and area through funnel $\frac{1}{8}$th of grate area.

Chapter VIII.

---

41. HANGER FOR SHAFTING. Plate XXXIX.

This form of bracket is used to support shafting from overhead beams.

42. WALL BRACKET FOR SHAFTING. Plate XL.

Sometimes shafting is required to be supported at a certain distance from the wall as well as from the floor, and in this case a *wall bracket* is used. The distance from the wall to the centre of the shaft is usually determined by the radius of the largest pulley intended to run on the shaft, and room is left to enable pulley and belt to be easily clear of the wall.

43. CYCLOIDAL AND INVOLUTE CURVES. Plate XLI.

The Cycloid.—When a circle rolls along on a straight line, a point in its circumference traces a curve called the cycloid. Thus (in Plate XLI.) suppose the circle, centre C, to roll upon the straight line A B. Then the path of the point P would describe the cycloidal curve shown. The curve may be constructed as follows:—Draw the line A B and the describing circle, centre C, 2 ins. diameter, touching it; also draw C D, the line of centres, parallel to A B. Set off from P on the line A B a distance P 12, equal in length to the circumference of the circle. This is obtained by multiplying the diameter of the circle by 3·1416; thus, the length P 12 will in this case be equal to $3{\cdot}1416 \times 2 = 6{\cdot}2832$ ins. ($6\frac{2}{7}$ ins.) Now divide the circumference of the circle into any convenient number of equal parts, say 12. Again, divide the length P 12 into the same number of divisions, and raise perpendiculars from each division to the line of centres C D. Then, if the circle be made to roll along on the line A B, the centre C would necessarily occupy the successive positions 1, 2, 3, &c., on the line C D; and the points 1, 2, 3, &c., on the circumference, would also coincide with the corresponding numbers on the line A B, until, after one complete revolution, P reaches position 12. These successive positions of the describing circle are shown by dotted arcs. When 1 on the circumference reaches 1 on line A B, the point P has moved up to point 1, and describes that part of the cycloidal curve from P to 1. The various points on the curve are obtained as follows:—Describe arcs of circles from centres 1, 2, 3, &c., on the line C D, and cut these arcs by horizontal

lines through points 1, 2, 3, &c., on the circumference of the circle. A line drawn freehand through the points of intersection gives the curve required. The cycloid is used to obtain the face of the teeth of a rack.

THE EPICYCLOID (*epi*, upon; *kuklos*, a circle).—When a circle rolls on the outside of another circle, a point on the circumference of the rolling circle will trace a curve called the epicycloid.

From centre R, with radius = 4 inches, draw the arc A B, upon the outside of which the circle centre C″ is to roll. From centre C″, with radius = $1\frac{1}{4}$ inches, draw a circle touching the arc A B. From centre R draw an arc C″D″ to represent the path of the centre of the circle as it rolls along. Join R C″. Let the point P, where R C″ cuts the arc A B, be the point whose path is to be determined. Divide the circumference of the describing circle, centre C″, into any convenient number of equal parts, say 12, and number them 1, 2, 3, &c. as shown. It is now necessary to cut off upon the arc A B a length equal to the circumference of the describing circle centre C″. Remembering that circumferences of circles are to each other as their radii; then since R P = 4 inches, and C P = $1\frac{1}{4}$ inches, length of arc P 12 on A B = $1\frac{1}{4} \div 4 = \frac{5}{16}$ of the full circumference of circle A B. Therefore angle P R 12 = $\frac{5}{16}$ of $360° = 112\frac{1}{2}°$. Having thus obtained the length P 12 on A B, divide it into 12 equal parts and number as shown. Then, as the circle rolls, 1, 2, 3, &c. on its circumference will coincide with 1, 2, 3, &c. on the arc A B. From centre R, draw lines through the divisions on arc A B, to cut the line of centres, and number the intersections as shown. These points are the successive positions of the centre of the circle. From them draw arcs of circles, with C″P radius, to represent the circle in its successive positions. From centre R draw arcs through the divisions on the circumference 1, 2, 3, &c., so that the arc through 1 shall intersect the arc struck from centre 1 on the line of centres, and so on. The line drawn through the points of intersection will give the epicycloid required. This curve is used to obtain the faces of the teeth of wheels.

THE HYPOCYCLOID (*hypo*, under; *kuklos*, a circle).—When a circle rolls upon the *inside* of another circle, a point in the circumference of the rolling circle will trace a curve called the hypocycloid.

The principle of its construction is similar to that of the epicycloid. P 12 on the arc A B is obtained as before. The successive centres of the circle, while rolling inside the arc A B, are found on the inner line of centres C D. Arcs drawn from centre R through the divisions on the circumference of the rolling circle, will intersect the circles drawn from 1, 2, 3, &c., on the line of centres C D; 1 intersecting 1, 2—2, &c. A line drawn through the points of intersection will give the hypocycloid required. This curve is used to form the flanks of the teeth of wheels. When the diameter of the rolling circle is *equal to the radius* of the circle within which it rolls, it will be found that the path traced by a point on the rolling circle becomes a straight line.

THE INVOLUTE of a circle is the name given to the path described by a point at the end of a string when the string is unwound from a cylinder. To describe the curve, draw the circle, centre C, 2 ins. diameter. Take any point P on its circumference, and draw a diameter P C 6, and from 6 draw a tangent to the circle. Make this line equal in length to half the circumference of the circle, that is, $2 \times 3{\cdot}1416 \div 2 = 3{\cdot}1416$ ins. ($3\frac{1}{7}$ ins.) Divide the line and the semicircle into the same number of equal parts, say 6. From each division on the circumference set out tangents (or lines at right angles to the line joining the point to the centre). Make 1 P = 6 5; 2 P = 6 4; 3 P = 6 3, &c. Join the extremities of these lines by a freehand curve. The path of the point is the involute of a circle.

In all the curves just described, the tangent at any point is perpendicular to the line joining that point with the point of contact of the rolling circle and fixed lines. Thus, in Plate XLI. the tangent *t t* at the point 5, for example, in any of the curves, is perpendicular to the line *a b*, joining the point in the curve with the corresponding point of contact of the rolling circle and fixed line. In other words, *b* is the centre and *a b* the radius of the circular path on which the point *a* in the curve tends to move at that instant.

44. WHEEL GEARING.

Toothed wheels are used to transmit rotary motion from one shaft to another. When the axes of the wheels are parallel, the wheels used are called *spur wheels*. (Plate XLIII.) When the axes of the wheels are not parallel, but intersect if produced, the wheels used are termed *Bevel Wheels*. (Plate XLIV.)

A wheel with a small number of teeth, especially when it gears with a larger wheel, is called a *pinion*.

A straight bar provided with teeth is called a *rack*.

The wheel which communicates its motion to the next is called the *driver*; that which receives the motion, the *follower* or *driven wheel*.

By the diameter of a toothed wheel is understood the diameter of the pitch circle.

The first step in drawing toothed wheels is to determine their respective *pitch circles*. These are imaginary circles intermediate between the tops and bottoms of the teeth (see Plate XLIII.), and may be considered to be the circumferences of a pair of *smooth rollers* in contact. Such rollers, however, could not usually be relied upon in practice to transmit motion merely by the friction at their two surfaces, hence teeth are added to the face of the rollers. Considering a pair of smooth rollers represented by the pitch circles rolling together by contact without slipping, it should be noted that each pair of points in the pitch surfaces which are in contact must be moving in the same direction and with the same velocity. Also, since the points in contact move in the same direction, the wheels themselves rotate in opposite directions (Fig. 72).

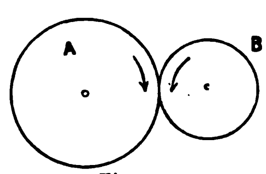


Fig. 72.

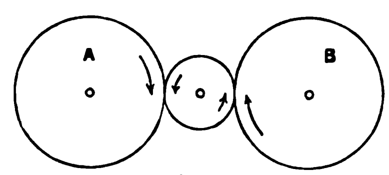


Fig. 73.

By adding an intermediate wheel (Fig. 73) the first and third wheels are made to rotate in the *same* direction.

Velocity Ratio.—We have said the points in contact at the surfaces of rollers A and B (Fig. 72) move with the *same* velocity; therefore, if the circumference of A is twice the circumference of B, while A makes *one* revolution B will make *two* revolutions. In such a case we should say the velocity ratio of A to B is as 1 : 2, or $\frac{A}{B} = \frac{1}{2}$. Hence, the number of revolutions made by a pair of

wheels is inversely proportional to their circumferences (or diameters, or radii); if A has 120 teeth, and B 40 teeth, since the teeth engage *one* by *one*, by the time 40 teeth of A have been engaged, B will have made one complete revolution, while A will only have made *one-third* of a revolution. Here the velocity ratio of A to B is as 1 : 3.

Or, if N = number of teeth in large wheel.
n = number of teeth in small wheel.
R = number of revolutions of large wheel per minute.
r = number of revolutions of small wheel per minute.

Then, since the number of revolutions is *inversely* as the number of teeth, N : n as r : R, or

$$\frac{N}{n} = \frac{r}{R}$$

Ex. **A wheel has 30 teeth, and makes 100 revolutions per minute; find the number of teeth for a wheel which is required to gear with it, and to make 40 revolutions.**

$$\text{Then } \frac{N}{n} = \frac{r}{R}\text{; or, } \frac{N}{30} = \frac{100}{40}\text{; } N = 75 \text{ teeth.}$$

Trains of Wheels.—When it is desired to obtain a high number of revolutions from a slowly rotating shaft, or *vice versâ*, use may be made of combinations of toothed wheels. If the combination of a large wheel on the driving shaft with a pinion on the driven shaft is not sufficient, intermediate shafts may be interposed having two wheels of unequal size on each, the larger wheel of one pair being made to gear with the smaller wheel of the next, thus:—

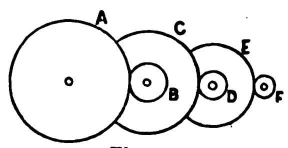


Fig. 74.

Let the number of teeth in the wheels, Fig. 74, be as follows: A = 100, B = 30, C = 80, D = 25, E = 60, F = 20; find the value of the train,

or, in other words, the velocity ratio, $\frac{A}{F}$.

$$\text{Then, velocity ratio} = \frac{\text{product of drivers}}{\text{product of driven}} = \frac{A \times C \times E}{B \times D \times F} = \frac{100 \times 80 \times 60}{30 \times 25 \times 20} = 32$$

That is, by the above arrangement of toothed gearing the number of revolutions per minute will be increased 32 times.

In Fig. 73 the intermediate wheel is called an *idle* wheel. Its effect is merely to change the *direction* of the wheels, and no matter how many intermediate wheels there may be, nor what their size, provided no two are on the same axis, they make no difference whatever to the relative velocity of A and B.

To draw the pitch circles of a pair of spur wheels having given their velocity ratio. Let A and B (Fig. 75) be the position of centres of axes, and let velocity ratio A : B be as 2 : 3. Join A B. From A draw A C three units long, and from B draw B D parallel to it on the opposite side of the line, and two units long. Join C D, cutting A B in E. Then A E : E B :: A C : B D, and A E and B E are the radii of the required pitch circles.

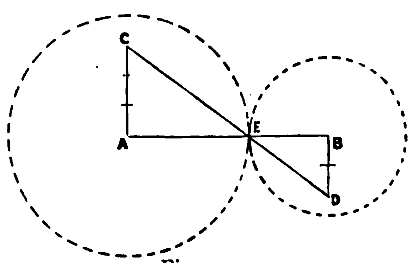


Fig. 75.

Pitch of Teeth.—The pitch of the teeth is the distance between a point on one tooth and the corresponding point on the next tooth to it, *measured on the pitch line.*

NOTE.—It has been said the pitch is measured on the pitch line. This is true, whether the pitch line is straight, as in the rack; or a circle, as in a wheel. In the circle the pitch is measured ***round the arc***, and not by the ***chord*** of the arc. Thus, suppose a wheel W and rack R to be represented by their pitch lines (Fig. 76), and suppose the wheel to roll along on the rack in the direction of the arrow; then, if $a\ b$ be the pitch measured on the straight line R b, the length $a\ b$ measured round the arc $a\ d\ c$ will be the pitch of the teeth of the wheel. For, as the wheel rolls, the point c should coincide with the point b, and this would not be the case if a straight line $a\ c$ were set off on the pitch circle equal to $a\ b$.

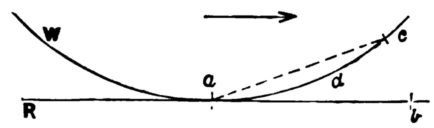


Fig. 76.

To lay off upon a circle an arc approximately equal to a straight line:—Let A C, Fig. 77, be part of the circumference of a circle, and let A B be drawn tangent to the circle and equal in length to the pitch. Make A D equal to ¼ A B. From centre D, with radius D B, describe the arc B C, cutting A C in C. Then A C is the arc required, equal in length to A B.

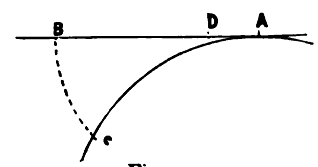


Fig. 77.

Having given the pitch circles and the number of teeth required, it is usually convenient, in order to obtain the centres of the teeth, to divide the pitch circles into the required number of equal parts, first by large divisions, and then each of these separately by subdivisions. Thus 20 teeth would be set out by dividing the circle first into 4 equal parts, and then dividing one of these quarters by trial into 5 equal parts. This will provide the distance required for the other subdivisions. Again, 30 teeth would be set out by dividing the pitch circle into 6 parts, by stepping round the circle with a distance equal to its radius, and then each sixth part into 5 parts. This is much better than trying to divide the circle by going all round it with small subdivisions.

The following equations give the relations between the pitch, diameter, and number of teeth:—

$$\text{(1.) Diameter of pitch circle} = \frac{\text{Pitch of teeth} \times \text{No. of teeth}}{3.1416}$$

$$\text{(2.) Pitch of teeth} = \frac{\text{Diameter of pitch circle} \times 3{\cdot}1416}{\text{No. of teeth}}$$

$$\text{(3.) Number of teeth} = \frac{\text{Diameter of pitch circle} \times 3{\cdot}1416}{\text{pitch of teeth}}$$

Ex. (*a*) Find the diameter of the pitch circle of a wheel with 60 teeth, ¾ in. pitch.

$$\text{By (1). Diameter} = \frac{{\cdot}75 \times 60}{3{\cdot}1416} = 14{\cdot}324 \text{ ins.}$$

The diameter of pitch circle may be set out from a decimal scale, or it may be expressed to the nearest $\frac{1}{16}$th or $\frac{1}{32}$nd of inch. Thus, 14·324 inches = $14\frac{5}{16}$ full.

Ex. (*b*) Find the pitch of the teeth in a wheel, the diameter of pitch circle being 14·324 ins., and number of teeth 60.

$$\text{By (2). Pitch} = \frac{14{\cdot}324 \times 3{\cdot}1416}{60} = {\cdot}75.$$

Ex. (*c*) Find the number of teeth for a wheel 14·324 ins. diameter, teeth ¾ in. pitch.

$$\text{By (3). Number of teeth} = \frac{14{\cdot}324 \times 3{\cdot}1416}{{\cdot}75} = 60.$$

The calculation of the diameter and pitch of toothed wheels is much facilitated by the following tables:—

Let P = pitch of teeth; D = diameter of pitch circle; N = number of teeth.

$$\text{Then, } N = \frac{D \times \pi}{P}; \qquad D = \frac{P \times N}{\pi}; \qquad P = \frac{D \times \pi}{N}$$

$$\text{Let } \frac{\pi}{P} = x; \qquad \frac{P}{\pi} = y; \qquad \frac{N}{\pi} = z$$

Then, number of teeth = D x; diameter of pitch circle = N y = P z; pitch of teeth = $\frac{D}{z}$

The following Tables give the values of x y z:—

TABLE XIII. VALUES OF x.

Pitch in inches.	0	⅛	¼	⅜	½	⅝	¾	⅞
0	—	25·1327	12·5664	8·3776	6·2832	5·0266	4·1888	3·5904
1	3·1416	2·7925	2·5133	2·2848	2 0944	1·9333	1·7952	1·6755
2	1·5708	1·4784	1·3963	1·3228	1·2566	1·1968	1·1424	1·0927
3	1·0472	1.0053	·9666	·9308	·8976	·8667	·8378	·8107
4	·7854	·7616	·7392	·7181	·6981	·6793	·6614	·6444

TABLE XIV. VALUES OF y.

Pitch in inches.	0	⅛	¼	⅜	½	⅝	¾	⅞
0	—	·0308	·0796	·1194	·1592	·1989	·2387	·2785
1	·3183	·3581	·3979	·4377	·4775	·5173	·5570	·5968
2	·6366	·6764	·7162	·7560	·7958	·8356	·8754	·9151
3	·9549	·9947	1·0345	1·0743	1·1141	1·1539	1·1937	1·2335
4	1·2732	1·3130	1·3528	1·3926	1·4324	1·4722	1·5120	1·5518

TABLE XV. VALUES OF z.

No. of teeth.	0	1	2	3	4	5	6	7	8	9
0	—	·3183	·6366	·9549	1·273	1·592	1·910	2·228	2·547	2·865
10	3·183	3·501	3·820	4·138	4·456	4·775	5·093	5·411	5·730	6·048
20	6·366	6·685	7·003	7·321	7·639	7·958	8·276	8·594	8·913	9·231
30	9·549	9·868	10·186	10·504	10 823	11·141	11·459	11·778	12·096	12·414
40	12·732	13·051	13·369	13·687	14·006	14·324	14·642	14·961	15·279	15·597
50	15·916	16·234	16·552	16·870	17·189	17·507	17·825	18·144	18·462	18·780
60	19·099	19·417	19·735	20·054	20·372	20·690	21·009	21·327	21·645	21·963
70	22·282	22·600	22·918	23·237	23·555	23·873	24·192	24·510	24·828	25·147
80	25·465	25·783	26·101	26·420	26·738	27 056	27·375	27·693	28·011	28·330
90	28·648	28·966	29·285	29·603	29·921	30·239	30·558	30·876	31·194	31·513

PROPORTIONS OF TEETH.—The following are particulars of proportions of teeth (Fig. 78):—*b d* or *e h* = thickness of tooth on pitch line; *b e* = width of space between teeth, *a b* = face of tooth, or part beyond pitch line; *b c* = flank of tooth, or part within pitch line; *d e* = side clearance

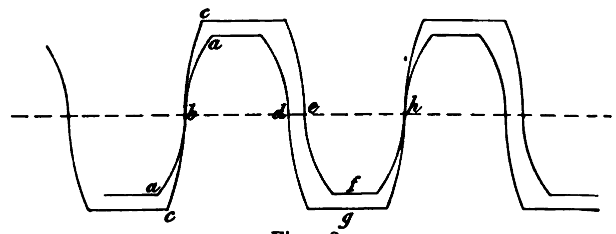


Fig. 78.

(that is, difference between thickness of tooth and width of space); *f g* = bottom clearance; *b h* = pitch.

Height above pitch line	*a b* =	$5\frac{1}{2}$	fifteenths of	pitch.
Depth below pitch line	*b c* =	$6\frac{1}{2}$	,,	,,
Thickness of tooth	*b d* =	7	,,	,,
Width of space	*d h* =	8	,,	,,

These proportions are conveniently obtained by drawing a line *a b* (Fig., 1 Pl. XLII.) equal to the pitch; then line *a c* of indefinite length, at any convenient angle. Mark off from *a* fifteen equal spaces, say $\frac{1}{8}$ths from the edge of a scale or rule. Join division 15 to *b*. From positions $5\frac{1}{2}$, $6\frac{1}{2}$, and 7, on line *a c*, draw lines parallel to 15 *b*, to meet the line *a b*. Then the pitch *a b* is divided as required.

Other proportions expressed in decimals in terms of the pitch *p* are:—

Height above pitch line	$a\ b = 0{\cdot}3\ p$
Depth below pitch line	$b\ c = 0{\cdot}4\ p$
Thickness of tooth	$b\ d = 0{\cdot}48\ p$
Width of space	$d\ h = 0{\cdot}52\ p$

The total height of the teeth, in small wheel drawings, may be taken as equal to $\frac{3}{4}$ths of the pitch.

Additional proportions for toothed wheels are:—

Thickness of rim of wheel = thickness of tooth on pitch line.
Thickness of arms, if flat, = thickness of tooth on pitch line.
Width of wheel for small pitches = pitch × 2.
Width of wheel for large pitches = pitch × 3.
Average width = pitch × $2\frac{1}{2}$.
Diameter of boss = diameter of shaft × 2.
Depth of boss = width of face of wheel × $1\frac{1}{4}$.
Breadth of arm at rim = pitch × $1\frac{3}{4}$, tapering toward boss.

These proportions may be somewhat modified according to circumstances, and according to the judgment and taste of the designer.

The diagram Fig. 79 gives the proportions of teeth for any pitch up to 4 ins. Thus suppose the pitch of teeth is $2\frac{1}{2}$ ins. On the horizontal line, which is marked in inches and eighths, find a

point *a* $2\frac{1}{2}$ ins. from the left hand end. Then on the vertical line from the point *a* take $a\ b =$ height above pitch line, $a\ c =$ depth below pitch line, $a\ d =$ thickness of tooth.

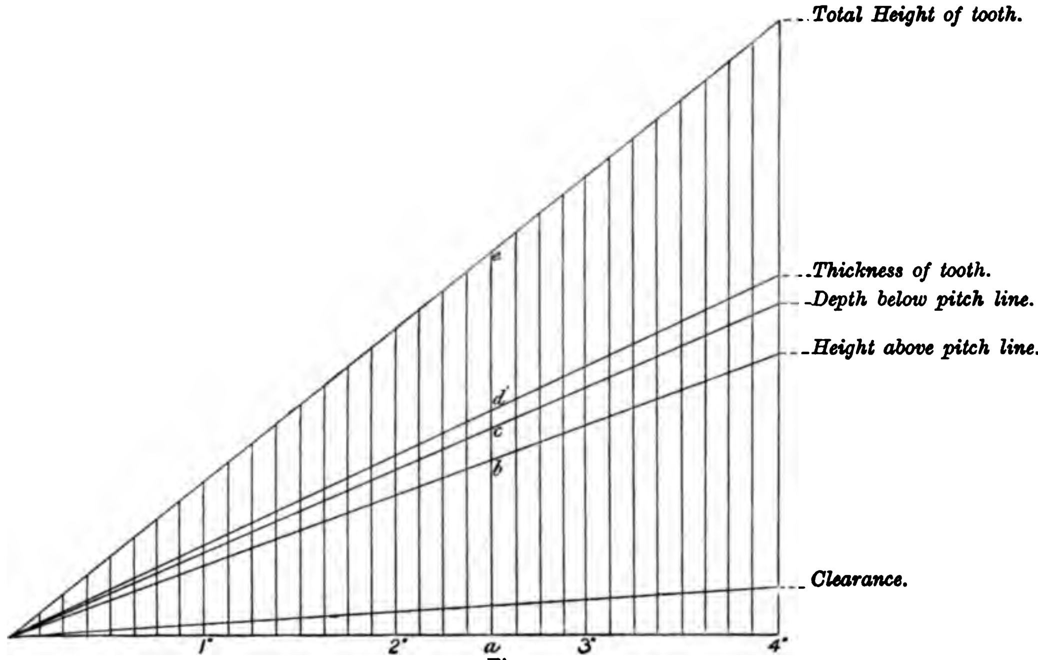


Fig. 79.

FORM OF TEETH.—Let a toothed wheel and rack as in Plate XLII. be in gear; then the motion of the wheel and rack relatively to one another is the same, whether the wheel rotates on

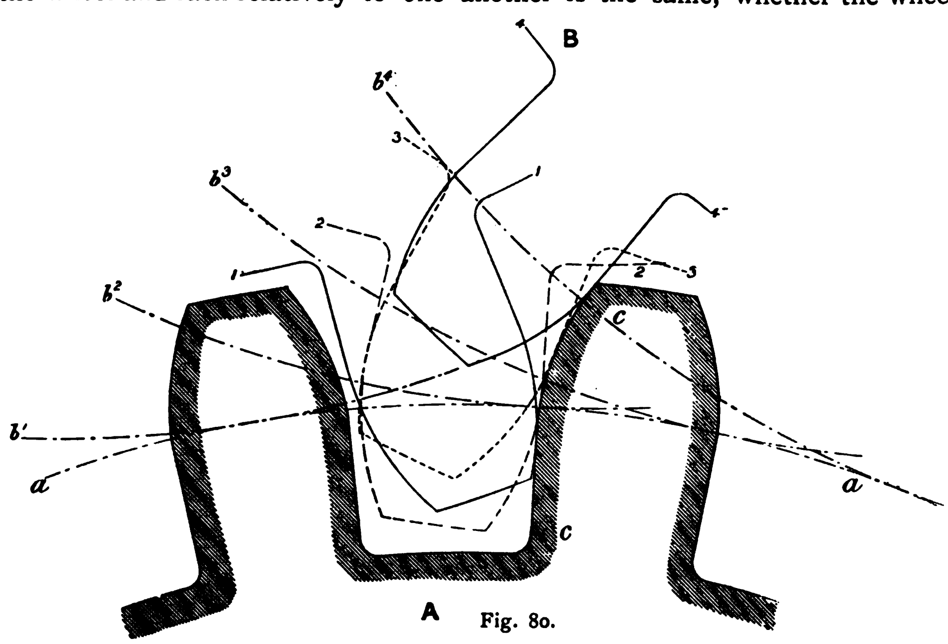


Fig. 80.

its fixed axis and the rack travels to and fro, or the rack is stationary while the wheel rolls from end to end upon it.

Again, let a pair of spur wheels, Plate XLIII., be in gear, and rotate on fixed axes; then the motion of the wheels relatively to one another is not changed if one wheel is fixed and the other rolls bodily round its circumference.

Referring to Fig. 80, let the pitch line b^1 of wheel B roll upon the pitch line *a* of wheel A, which remains stationary; then in order that a tooth of B of given shape may roll smoothly and be continuously in contact with the tooth of A, it is evident that the curve of the tooth *c c* must be tangent to the successive positions b^1, b^2, b^3, b^4, &c., of the tooth of B, and a correct form for the tooth of A may be obtained by tracing it to touch the tooth of B in its successive positions.

The form of teeth thus obtained would be such as to give by their sliding contact the same perfect uniformity and smoothness of motion as would be obtained by the rolling together of a pair of smooth rollers, represented by the pitch circles; and it is shown in works on Applied Mechanics that if the acting surfaces of the teeth be formed of *cycloidal* or *involute* curves, they fulfil these required conditions perfectly.

The cycloidal curve is the one most frequently used, and we will now illustrate its application in obtaining the correct form of wheel teeth. Let A and B (Fig. 81) be the pitch circles of a pair of wheels in gear, and let C be the describing circle, which may be taken of any convenient size, in contact with both wheels at the same point E, and which is used to obtain the profiles of the teeth. Now suppose all the three wheels to rotate together on their respective fixed axes, without slipping,

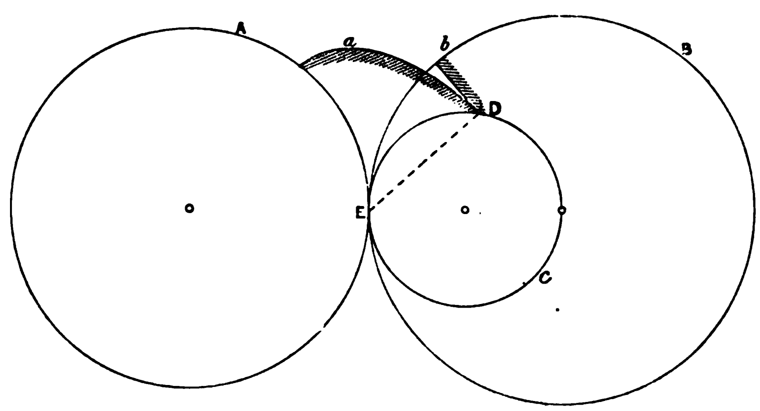


Fig. 81.

and suppose at point D, on wheel C, a pencil is secured which will describe curves as the wheels rotate. Let the pencil D describe the epicycloid *a* on a sheet of paper *attached to and rotating with* A; and at the same time let it mark the hypocycloid *b* on wheel B, both curves having been commenced by the pencil D at the common point E. The curve *a* is the same as would be formed by the pencil D if A were fixed and C made to roll within it. The curve *b* is the same as would be formed by the pencil if B were fixed and C made to roll within it.

If now these curves are used for the profiles of two teeth gearing together, they will remain in contact, for they have both been described at the same time by the same point; and in a pair of wheels with teeth so formed, if one be used to drive the other the teeth will slide the one on the other, and maintain the same uniform angular velocity as the pitch circles of the wheels.

The curves in Fig. 81 touch one another in a common point D, and they have D E *a common perpendicular to the surfaces in contact at that point, which will invariably pass through the point of contact of the pitch circles;* and this is the sole condition necessary to ensure a constant velocity ratio of the two wheels.

In practice these curves may either be drawn as in Plate XLI., or they may be constructed by rolling a disc (or part of a disc) of wood representing the "describing" circle, on inside and outside templets, also made of wood and struck out to the radius of the pitch circle. A needle or sharp-pointed wire is pushed through the wooden disc, so that its point appears at the edge and coincides with the circumference. Care is taken to avoid slipping, and the required curve is scratched upon the wood or paper upon which the templets are laid.

The shaded parts on Fig. 82 show the form in which the templets may be made. A careful draughtsman may obtain the curves by rolling along the pitch circle a stiff paper templet, and setting off equal distances on the arc of the templet and of the pitch circle. By carefully making the divisions to coincide as the paper rolls, and marking the position of the point whose path is to be traced with a pencil, the required curve may be obtained.

It will be clear by reference to any of the diagrams that the face of one tooth gears only with the flank of its fellow; and therefore the curves described, as shown on Fig. 81, represent the face of one and the flank of the other. An entirely separate operation, by a similar method, will be necessary for obtaining the remaining portion of each tooth. Hence *two* describing circles are required in each case.

THE SIZE OF THE DESCRIBING CIRCLE.—The diameter of the describing circles for a pair of wheels gearing together is arbitrary, and is chosen according to the judgment of the designer. They may be both the same diameter, or they may be of unequal diameters, providing that the same circle be used to describe those parts of the teeth which gear together, namely, the *faces* of the teeth of one wheel with the *flanks* of the teeth of the other wheel, also that the diameter of the describing circle be not greater than the radius of the pitch circle in which it rolls.

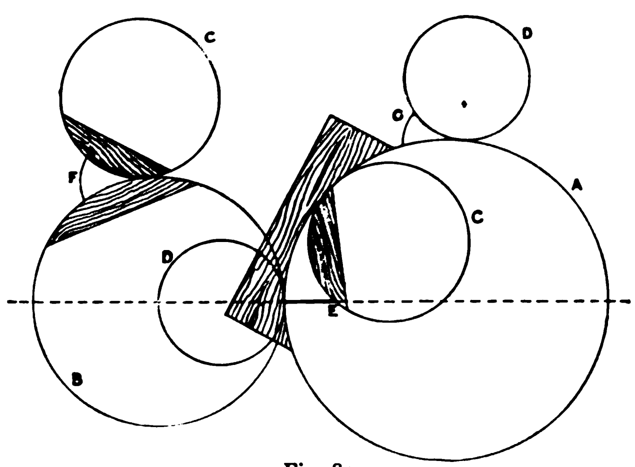


Fig. 82.

Let A and B (Fig. 82) be the pitch circles of a pair of wheels upon which we are to describe teeth, of the cycloidal form, which will work smoothly together. The part of the tooth *above* the pitch line on one wheel works with the part of the tooth *below* the pitch line on the other wheel of the pair, and those parts which work together are formed with the same describing circle. Hence, if C be the describing circle chosen for the flanks of the teeth of wheel A, the same circle must be used to describe the faces of the teeth of wheel B, and *vice versâ*; if D describes the flanks of the teeth for wheel B, it must also be used to describe the faces of the teeth of A. Since the diameter chosen for C is one-half the diameter of A, the path of a point on C rolling inside A will be a *straight line* E passing through the centre of A; and this will form the radial flank of a tooth as shown in the Plates.

The same circle C, rolled on the outside of B, gives the curve F, which forms the *faces* of the teeth of B to gear with the *flanks* of the teeth of A.

Similarly the describing circle D, being half the diameter of B, gives radial flanks on B, and it is used to form the curve G for the faces of the teeth on A.

When a *set* of several wheels are to be constructed so as to be interchangeable, the *same* describing circle must be used for the faces and flanks of *every wheel* of the set. It is then desirable to take for the diameter of the describing circle the radius of the smallest wheel of the set.

If the diameter of the describing circle be half the diameter of the pitch circle, as we have already seen, the flanks of the teeth will be radial lines. (Fig. 83 A.) If the diameter of the describing circle be *less* than half the diameter of the pitch circle, the teeth will be curved outwards from the pitch line downwards, as in Fig. 83 B, thus considerably strengthening the root of the tooth. Lastly, if the describing circle be *greater* than half the pitch circle, the effect is to reduce the thickness of the tooth at the root, as in Fig. 83 C.

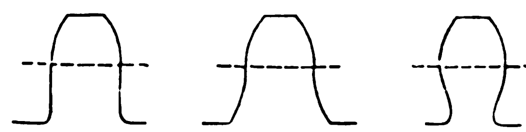


Fig. 83.

This latter case is most objectionable; hence the diameter of the describing circle should never be greater than half the diameter of the pitch circle of the wheel.

Path of Contact.—Since the parts of the teeth in contact have been described by a common point on their respective describing circles, the *path of contact* of the teeth takes place along those circles, namely, along the double arc *e p c*, Fig. 84, *e* and *c* being the points of intersection of the arcs passing through the tops of the teeth with the describing circles *s* and *r*. If radii be drawn from the centre of the pitch circle to the points *g* and *f* where the contact of a pair of teeth begins and ends, they will cut off an arc on the pitch circle, termed the *arc of contact*.

If B be the driving wheel, then the arc *g p* of the pitch circle is called the *arc of approach*, and the arc *p f* is called the *arc of recess*. At least one pair of teeth must be in gear at one time, hence the arc of contact should be greater than the pitch. In ordinary gearing the arc of contact varies from 1·6 to 2 times the pitch.

Wear and Tear.—One toothed wheel drives another by virtue of the *sliding action* of the teeth of one wheel upon those of another. This can be clearly seen in the case of the Rack and Pinion (Plate XLII.); the tooth of the Pinion, which is in contact with the rack at *s*, slides upon it along its whole face from *s* to *p*, until contact between the two teeth ceases.

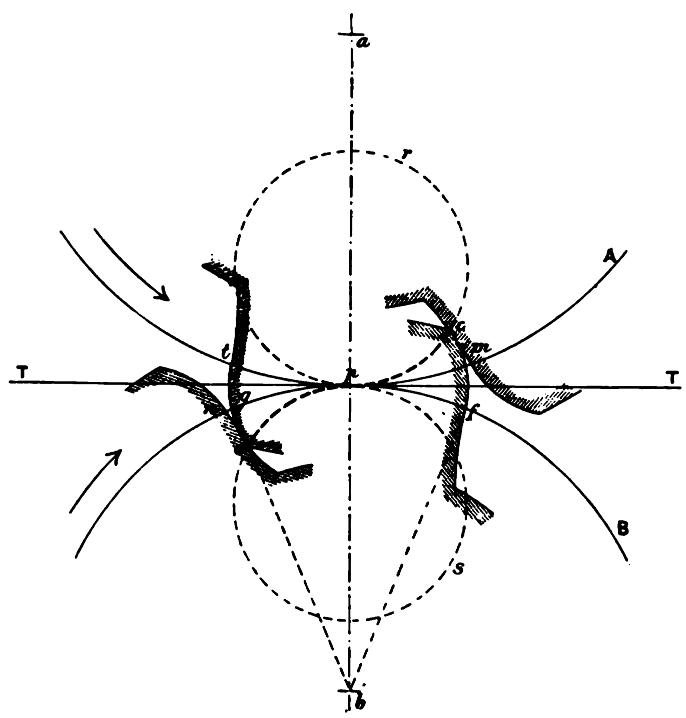


Fig. 84.

Referring to Fig. 84, the face *t e* of the tooth of the driven wheel A slides upon the flank *n e* of the tooth of the driver B, until *t* and *n* coincide at the pitch point *p*. The amount of sliding of the face relatively to the flank is therefore equal to *t e* — *n e* (because *t* does not slide to *e* but only to *n*).

Now the wear and tear on the flank of the tooth of B is distributed over the surface *n e*. But it should be observed the length *n e* is governed by the diameter of the describing circle; for with a larger circle the point of intersection *e* would approach nearer *n*, and make the distance *n e*, over which the wear and tear occurs, shorter. Hence the effect of a ***small*** describing circle is to distribute the wear and tear on the flank over a greater length of surface. On the other hand, a small describing circle reduces the length of the path of contact.

If the point *e* (Fig. 84) be joined to *p* by a straight line *e p*, it makes an angle with T T the common tangent to the pitch circles through *p*. This angle is called the ***angle of obliquity*** of action of the teeth at the point *e*, and is greatest at the commencement and the end of contact; while at the instant the teeth pass the line of centres at *p*, the angle of obliquity is nothing. This angle also varies with the size of the describing circle, being greater with a small than with a large describing circle

The effect of reducing the size of the describing circle may be summarised thus :—

1. It increases the thickness of the tooth at the root.
2. It distributes the wear and tear on the flank of a tooth over a greater length of surface.
3. It reduces the length of the path of contact, and reduces, therefore, the number of points of contact, or of teeth in contact, at one time.
4. It increases the obliquity of action of the teeth.

STRENGTH OF WHEEL TEETH.—The dimensions of the parts of a toothed wheel depend upon the power to be transmitted through the teeth. But the power transmitted, or the work done at the rim of the wheel, expressed in horses' power, is the product of the pressure on the teeth multiplied by the velocity of the rim per minute. Hence for a given power the load on the rim varies inversely as its velocity. In proportioning wheels, therefore, to transmit a given power, the lower the velocity of the rim the greater must be the strength of the parts. The velocity of the rim depends of course upon the diameter of the wheel as well as upon the revolutions of the shaft upon which it is placed.

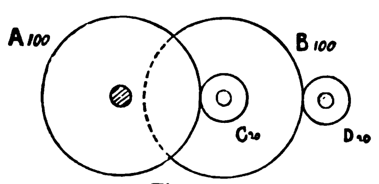


Fig. 85.

Thus, considering a simple train of wheels (Fig. 85) the ratio of the revolutions of A and D as before explained (page 106), is A : D :: 1 : 25; but the velocities of the rims of these wheels is not expressed by this ratio, for velocity of rim of A is equal to that of C; but velocity of rim of B is five times that of C, and therefore also five times that of A; but velocity of rims of B and D are also equal; therefore velocity of rim of D is five times that of A; or, pressure on teeth of D equals one-fifth that on teeth of A. The same result may be obtained by the principle of the lever.

To find the pressure P on the teeth of wheels, having given the horse-power H transmitted through the wheel; the number of revolutions N of the wheel per minute; and the radius R of the wheel in feet.

Then the pressure in lbs. at the rim of the wheel multiplied by the velocity (V) of the rim in feet per minute, will give the number of foot lbs. of work done at the rim per minute; and this product

must equal the number of horses' power transmitted through the wheel, also expressed in foot lbs. Therefore

$$P \times V = H \times 33000;$$
$$\text{or, } P = \frac{H \times 33000}{V}$$
$$= \frac{H \times 33000}{2\pi R \times N}$$
$$= 5252 \frac{H}{R\,N}$$

Ex. Find the pressure transmitted through the teeth of a spur wheel which transmits 70 horse-power; number of revolutions of wheel per minute = 60; radius of wheel, 1·5 ft.

Then, by above equation

$$P = 5252 \frac{H}{R\,N}$$
$$= \frac{5252 \times 70}{1{\cdot}5 \times 60}$$
$$= 4085 \text{ lbs.}$$

The teeth of wheels are not always made with sufficient accuracy to ensure that when a pair of teeth are in contact they bear along the whole width of the tooth, and they are therefore designed so as to be strong enough to resist fracture when the whole load to be transmitted is carried by one corner of one tooth.

Fig. 86 represents a section of part of the rim of a wheel, with a longitudinal view of the tooth $a\,c\,e\,f$. If the load P acts on the corner c instead of being carried by the whole width of the tooth, then the tooth tends to fracture along some line $a\,n$. Draw $c\,m$ perpendicular to $a\,n$; and let $a\,n = b$, and $c\,m = l$; and let $d =$ the thickness of the tooth. Then the bending moment of the force P acting at c about $a\,n = P \times c\,m = P \times l$. The moment of resistance at the section through $a\,n = \frac{f\,b\,d^2}{6}$ (page 21). Then the section is sufficiently strong when $P \times l = \frac{f\,b\,d^2}{6}$

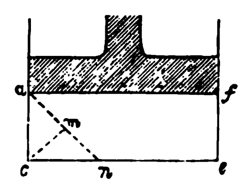


Fig. 86.

where f is the safe working load = 4,500 lbs. for cast-iron.

The stress at the section is greatest when the angle $c\,a\,n = 45°$, and $b = 2\,l$.

Then the thickness of the tooth d, which may be obtained from the above equation, thus:—

$$d^2 = \frac{6\,P\,l}{f\,b}$$

now becomes

$$d = \sqrt{\frac{3\,P}{f}}$$

or, in words, to find the thickness of the teeth of a cast-iron wheel to carry a given load, taking $f = 4{,}500$, divide the load by 1,500 and extract the square root.

Ex. Find the thickness of the teeth of a wheel to carry a load P of 1 ton = 2240 lbs., taking $f = 4500$ lbs. Then, by formula,

$$\text{Thickness, } d = \sqrt{\frac{3 \times 2240}{4500}}$$
$$= 1\tfrac{1}{4} \text{ ins. nearly.}$$

16

Assuming that the load is distributed over the whole width of the tooth, which is the more usual condition in well-constructed gearing, let the tooth of wheel A as driver act on the tooth of B as shown, Fig. 87. Then the tooth of B may be considered as a cantilever fixed at one end and loaded at the other, and its strength may be calculated by the ordinary rules for rectangular beams.

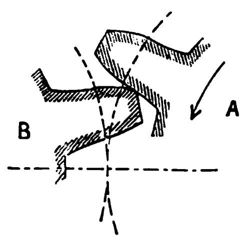


Fig. 87.

Ex. Find the breaking load P for cast-iron teeth (Fig. 88):—Length, $l = 2$ ins.; breadth, $b = 8$ ins.; depth, $d = 1{\cdot}5$ in.

Now a bar of cast-iron 1 inch square and 1 inch long, when secured at one end, breaks with a load of 6,000 lbs. at the other end. Then, since the strength of rectangular beams is directly as their breadth, as the square of their depth and inversely as their length, we have the breaking weight P from the following equation:—

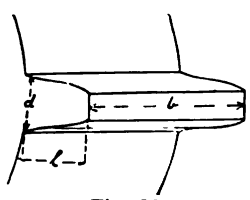


Fig. 88.

$$P = \frac{b \times d^2 \times 6000}{l}$$

$$= \frac{8 \times 1{\cdot}5 \times 1{\cdot}5 \times 6000}{2}$$

$$= 54000 \text{ lbs.}$$

If we divide the breaking load by the factor of safety (which for wheel gearing may be taken as 10) we have for the safe working load $54{,}000 \div 10 = 5{,}400$ lbs.

This example assumes that the whole load is carried by one tooth. If, as is usually the case, there are at least two pairs of teeth in gear at one time, the working load for the wheel is equal to 5,400 multiplied by some factor between 1 and 2, say 1·5.

The pressure P which may be transmitted through wood teeth in mortise gearing may be taken $= 0{\cdot}7$ that for iron teeth.

45. RACK AND PINION. Plate XLII.

To draw this example, draw first the pitch line of the rack, which will be an indefinite straight line; then the pitch circle of the pinion, $6\frac{3}{8}$ ins. radius, *touching* the pitch line of the rack; and then the vertical centre line.

The first thing we require to know is the *pitch* of the teeth. This is given on the drawing, namely 4 ins. On each side of the point *s* where the two pitch surfaces touch, namely, at the intersection of C D and the pitch line of rack, we set off the thickness of the tooth *s r* on the pitch circle of pinion, and *s t* on the pitch line of rack.

This must be obtained by dividing the pitch into fifteen parts by the method before explained, and taking seven of these parts for the thickness *s r* and *s t*, or from the figure, page 110.

If we determine that the flanks of the pinion teeth shall be radial, that form will of course be given by a describing circle whose diameter equals the radius of the pitch circle. Hence this circle (centre K) rolled upon the pitch line of the rack, will describe the faces of the rack teeth. The construction of the cycloidal curve *e f* by the method previously explained is shown. We may now find, by trial, a centre from which to strike an arc approximately coinciding with the curve; or, better, a templet may be made to the exact form.

Again, if we decide that the flanks of the teeth of the rack shall be radial, assuming that the pitch line of the rack is the circumference of an infinitely large circle, we have a describing circle also in-

finitely large; hence the flanks of the rack teeth are drawn by parallel lines perpendicular to the pitch line. Now, using this describing circle—which is, in effect, a straight line—for the faces of the pinion teeth, we describe upon the pitch circle the involute of the circle *g h*. (See also Plate XLI.) This curve may now be drawn for the remainder of the teeth with an approximate radius of $3\frac{1}{4}$ ins. from a line of centres which in this case will be found just outside the pitch circle.

46. WHEEL AND PINION IN GEAR. Plate XLIII.

Plate XLIII. shows a wheel and pinion in gear. The wheel has 32 teeth, pitch of teeth $\frac{7}{8}$ in. Thus $32 \times \frac{7}{8} \div 3\cdot1416 = 8\cdot912 = 8\frac{29}{32}$ ins. diameter. (See page 108.) The pinion has 20 teeth, pitch $\frac{7}{8}$ in.; therefore diameter $= 20 \times \frac{7}{8} \div 3\cdot1416 = 5\cdot57 = 5\frac{9}{16}$ ins. nearly.

First, therefore, draw a horizontal centre line, and set off upon it two circles (which will be the pitch circles) of the above diameter, touching one another.

Draw vertical centre lines, thus dividing each pitch circle into four equal arcs. Divide arc *a b* of the wheel into eight equal parts. This will require care and accuracy on the part of the student.

These divisions, having been correctly obtained for one quadrant, may be used to divide the remainder of the circle. (If they do not come true, your centre lines are not square.) Repeat the process for the pinion, dividing the arc *a c* by trial into five equal parts, and then completing the divisions throughout.

Now set off on each side of *a* the thickness of a tooth on the respective pitch circles (obtained by taking 7 fifteenths of the pitch, as before explained), and repeat this from every division already obtained. Mark distances $5\frac{1}{2}$ fifteenths of the pitch outside the pitch circles, and $6\frac{1}{2}$ fifteenths inside the pitch circles, and draw circles through these points for the tops and bottoms of the teeth.

Assuming that we wish the flanks of the teeth to be radial, draw lines from the divisions on the pitch circles towards their centres—conveniently done by putting a needle in the centre of the circle and keeping the straight edge against it. Notice that the radial flanks are rounded a little at the bottom to strengthen them. As before explained, radial flanks must have been described by a circle whose diameter is half that of the circle inside which it rolls. The describing circles *d* and *e* are shown dotted; *d* describes the flanks of the wheel teeth and the faces of the pinion teeth; *e* the flanks of the pinion teeth and the faces of the wheel teeth. The face, as at *c*, may be obtained by constructing the curve geometrically, or by rolling a cardboard templet. (See Fig. 82.) The remainder of the figure explains itself.

The lower view shows half of the wheel and pinion in section. The lines across it represent the edge view of the teeth, and they are obtained by projecting from the top corners of the teeth of each wheel.

47. BEVEL WHEELS. Plate XLIV.

When the axes of a pair of shafts are not parallel, bevel wheels are used to transmit rotary motion from one to the other.

(1.) Let the axes be at right angles and in the same plane. Let A (Fig. 89) be the driver and B the follower, and let the velocity ratio $\frac{A}{B}$ equal $\frac{2}{1}$; that is, A makes two revolutions to one of B.

Then the wheels are struck out as follows:—

Draw centre lines of shafts (Fig. 89) *a a* and *b b* at right angles, intersecting in *c*. Mark off on *c b* one unit of length (any distance) and on *c a* two units, and complete the rectangle 2 *d* 1 *c*. Then supposing the arrangement to be a pair of smooth cones rolling upon one another, and having a common apex in *c*, the line of contact of the cones is in *c d*. The position of the point *g* on *c d* depends upon whether the wheels are required to be large or small. Let *e g* be the required radius of wheel A.

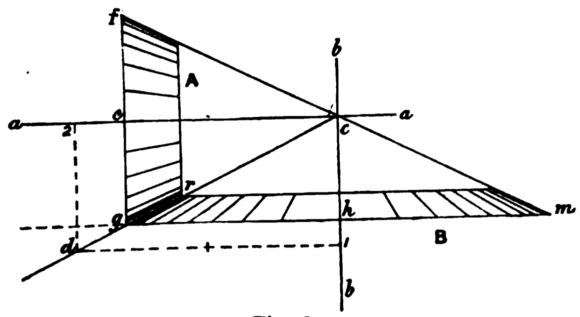


Fig. 89.

Draw a line parallel to *a a*, and at a distance *e g* from it, cutting *c d* in *g*. Then *e g* is the radius of wheel A, and *g h* drawn parallel to *a a* is the radius of wheel B. For *e g* : *g h* :: 1 : 2, which is the ratio required. Make *e f* equal to *e g* and *h m* equal to *h g*, and draw *f c m*, completing the contour of the cones. Parts only, or frusta, of these cones are used to form the wheels. Thus the part *g r* is measured along *c d* to the length required for the teeth of the wheels.

When the axes of a pair of wheels are at right angles, and their diameters are *equal*, they are called *mitre wheels*.

(2.) When the inclination of the axes of the shafts is not a right angle, let the velocity ratio $\frac{A}{B}$ equal $\frac{2}{1}$ as before, using the same lettering, and proceeding as in the last example.

Draw the centre lines of shafts *a a* and *b b* (Fig. 90) to the inclination required (say 60°), intersecting in *c*. Mark off on *c a* two units of length, and on *c b* one unit. Complete the parallelogram as before by drawing lines from the extremities of the division parallel to the axes of the shafts and intersecting in *d*. Then *c d* is the line of contact.

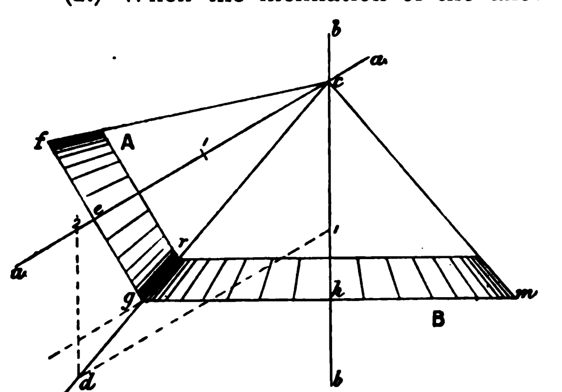


Fig. 90.

Let *e g* be the required radius of wheel A. Draw a line parallel to *a a* at a distance *e g* from it, cutting *c d* in *g*. Through *g* draw lines *g e f* and *g h m* perpendicular to the axes *a a* and *b b* respectively. Make *e f* equal *e g*, and *h m* equal *h g*. Join *c f*, *c m*. Frusta of the cones are taken, length *g r*, of which to form the wheels.

In the Plate XLIV. three bevel wheels A, B and C are shown in gear. Wheel A (Fig. 1) is shown in section, and in gear with wheel C, which is shown in half-section. The remainder of the view shows the teeth of wheels B and C complete.

Draw the centre lines of the shafts intersecting in *o′*.

Wheels A and B have 24 teeth, $1\frac{5}{8}$ pitch; therefore $24 \times 1\frac{5}{8} \div 3{\cdot}1416 = 12{\cdot}414 = 12\frac{7}{16}$ diameter nearly.

Wheel C has 20 teeth; $20 \times 1\frac{5}{8} \div 3{\cdot}1416 = 10{\cdot}345 = 10\frac{3}{8}$ diameter nearly.

Set off the lines *a b* and *d c* $10\frac{3}{8}$ ins. apart, at an equal distance on each side of the centre line of wheel C (Fig. 1), and *a d* and *b c* $12\frac{7}{16}$ ins. apart. The lines *a b*, *b c*, and *c d* are the projections of the pitch circles of the wheels.

Draw *a o' c*, *b o' d*. From *c* draw *f c e* at right angles to *o' c*. Draw also at *a*, *b* and *d* lines at right angles to *a o'*, *b o'*, *d o'*, and produce them as shown. At *d*, on the line *d e*, make *d t* and *d z* equal to the height of the tooth above and depth below pitch line, as in the spur wheel drawing. This is done by dividing the pitch into 15 parts in the usual way, and taking $5\frac{1}{2}$ parts for *d t*, and $6\frac{1}{2}$ parts for *d z*. Set off similar marks for height and depth of tooth from *a*, *b* and *c* for each of the wheels. It will be noticed that at *b* we mark off four points, viz., the top and bottom of the tooth of A, and the top and bottom of the tooth of C. The remainder of the sectional parts of the wheels may now be completed from the dimensions given, care being taken to notice that the lines of teeth and rim radiate to the centre *o'*. We will now proceed to obtain the correct shape of the teeth. Referring to wheel B, we have *d c* the projection of the pitch circle, which we may consider also as the projection of the common base of two cones *d o' c* and *d e c*; and it is upon the surface of the cone *d e c* that the shape of the teeth of wheel B is delineated. Similarly, it is upon the surface of the cone *f c b* that the shape of the teeth of C is drawn. In order, therefore, to obtain the true form of the teeth, a small portion of the cones is shown unwound, or "developed," in Fig. 2. To prevent crowding of the lines the centres *f* and *e*, which are the apices of the cones, are removed to any convenient distance *k* and *g*. From centre *k*, with radius *f c*, the arc *l* is drawn; and from centre *g*, with radius *e c*, the arc *p* is drawn, touching the arc *l*; and upon these arcs, which are the development of the pitch circles, the true shape of the teeth is shown. The flanks of the teeth in both wheels are made radial, and the describing circles *n* and *m* rolling upon the arcs *l* and *p* give the form of the teeth required.

Fig. 3 is now drawn commencing with the pitch circle, with radius $o\ a =$ half *c d* or *b a*. Then set off on the pitch circle the centres of the teeth $1\frac{5}{8}$ in. pitch. The height *z t* in Fig. 3 is not the real but only the apparent height of the tooth. Its real height is given at *z t*, Fig 1; it is also shown in Fig. 2. To obtain *t* and *z* in Fig. 3, project from the corresponding points in Fig. 1. Now draw circles to represent the tops and bottoms of the teeth. The thickness of the teeth on the pitch line, also at the top of the tooth of Fig. 3, is obtained from Fig. 2, and these thicknesses are set off on each side of the centre line of each tooth. The roots of the teeth radiate to the centre *o*.

The top half of the Fig. 3 shows the back of the wheel, and the bottom half the front of the wheel. So far as we have gone the shape of the teeth is the same all round the rim. For the front view of the teeth of wheel B, Fig. 3, draw circles for the tops and bottoms of the teeth at the smaller end. Thus from centre *o*, radius *o h*, Fig. 3 (equal to *i h* Fig. 1), describe the circle for the tops. Similarly *i s*, Fig. 1, gives the radius for the bottoms. The straight lines are drawn radiating to the centre *o* as shown, and these lines determine the thickness of the small end of tooth.

For the teeth of wheel B, Fig. 1, the vertical lines all correspond with similar vertical lines on the sectional view. The tops *t* are obtained by projecting from Fig. 3 on to the vertical line through *t*, Fig. 1. Again, the thicknesses of the teeth on the pitch line through *d* are projected from the pitch line of the corresponding teeth on Fig. 3. The roots of the teeth may also be projected from Fig. 3, or they may be drawn by lines radiating from the pitch line towards *e*, or for the smaller ends towards *r*. Lines representing the width of the teeth all radiate towards the centre *o*.

To obtain the side view of the wheel C, Fig. 3, Fig. 4 has been constructed by the methods already explained, and the points required for the side view are obtained by projecting as before. The teeth of wheel C shown in Fig. 1 are transferred from the same wheel on Fig. 3. In cases where such transfer is not possible owing to an odd number of teeth, a separate plan of wheel C would have to be drawn under Fig. 1.

48. INVOLUTE TEETH.

The simplest form of tooth is the involute tooth, which is formed by one continuous curve for both face and flank. Having given the pitch circles *a* and *b*, Fig. 91, of two wheels in gear, we draw through the pitch point *c* a line *p r* at any desired angle representing the path of contact of the teeth, or the line of action of the mutual pressure between the teeth. The angle which this line makes with the common tangent *t t* to the pitch circles is called the "angle of obliquity" of the line of action, and it should not exceed 15°.

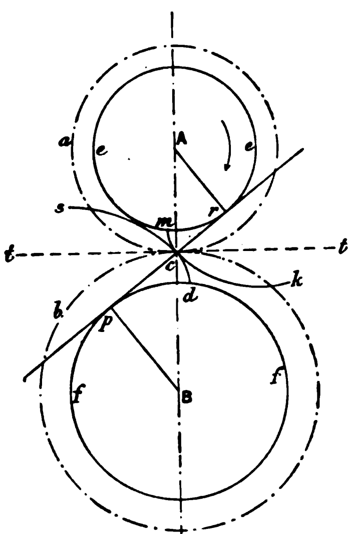


Fig. 91.

From centres A and B draw perpendiculars A *r*, B *p*, to the line *p r*, and with these lines as radii draw the circles *e e* and *f f*, called the *base circles* (B C).

Draw involutes *d c s*, *m c k* of the base circles *e e* and *f f*, and let them be in contact at *c*. Let wheel A drive B by the sliding contact of the involutes; then it can be shown that the wheels will rotate with a constant velocity ratio.

The details of drawing the teeth are illustrated by Fig. 92. In the example chosen we have a pair of wheels 35 teeth, ¾ in. pitch; diameter of pitch circles, 8⅜ ins.; angle of obliquity, 15°. The pitch circles touch at the point *c*, the angle of obliquity is taken at 15° and Q Q is drawn making 15° with a horizontal line through *c*, and the base circles are drawn with radius A *s*, that line being perpendicular to Q Q. From the pitch point C, set off the pitch round the pitch circles. With the usual proportions (page 109) set off the thickness of tooth on the pitch line, and the height of tooth above and depth below pitch line, and draw circles for the tops and bottoms of the teeth, called respectively the *addendum* circle and the *root* circle. The addendum circles cut the line of contact Q Q in *m* and *n*. With A as centre and radius A *n*, and from B as centre, radius B *m*, draw the *flank circles* F C, shown dotted in the figure just beyond the base circles. These circles give the lowest points of contact on the flanks of the teeth.

At any convenient point on the *base circles* draw involutes of those circles for the curves of the teeth as shown at V, and find an approximate arc of a circle to fit these curves; then with the arc obtained describe that part of each tooth between the addendum and flank circles. For method of describing the involute see page 104.

The remaining portions of the teeth between the flank circle and the bottom or *root* circle are drawn tangent to the involute. To draw the tangent, from centre A with radius A *t* = *n s*, the difference between *c s* and *c n*, describe a circle; then *t n* is a tangent to the involute at point *n*; and all the teeth may be completed from the base circle to the root circle by drawing lines to touch the circle with radius A *t*. Of course the two tangent lines of each tooth should not cross each other. The bottom corners of the teeth should be rounded as shown.

The length of the path of contact is *m n*, being the part of the line Q Q which is cut off by the outer or addendum circles of the respective wheels. These intersections show where contact begins and ends. Thus, if the upper wheel A is the driver, and rotates in the direction

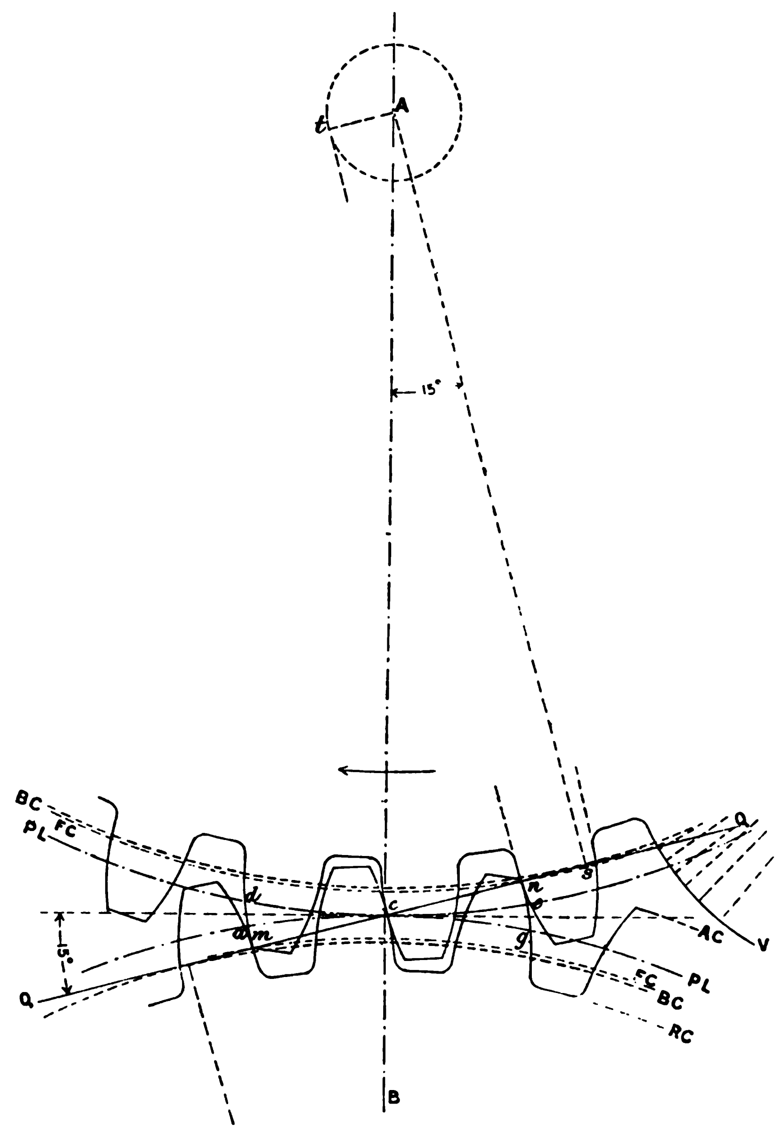


Fig. 92.

of the arrow, contact begins at *n* and ends at *m*; *n e* is the length of the flank of teeth of A, and *n g* the length of the face of those of B; *n e* and *n g* being the parts of the respective teeth which come in contact during approaching contact. Similarly, during receding contact, *d m* is the length of the face of the tooth of A, and *a m* the length of the flank of the tooth of B, these being the parts of the respective teeth which slide on one another.

In order that there may be contact between two pairs of teeth at one time, the length of the path of contact *m n* must not be less than twice the normal pitch. The ***normal pitch*** is the distance measured along the line of contact from the front of one tooth to the front of the next. It may be obtained as follows:—Let *c f* (Fig. 93) be the tangent to the two pitch circles drawn through the pitch point *c*. Make $c\ b =$ the pitch $=$ in the present example $\frac{3}{4}$ inch. Let $f\ c\ Q =$ the angle of obliquity. From *b* draw *b n* perpendicular to *c* Q. Then *c n* is the ***normal pitch***.

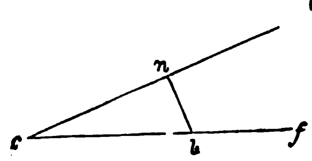


Fig. 93.

With an angle of obliquity of 15° the smallest number of involute teeth in a wheel should be twenty-five, in which case $c\ n = c\ s$, and the tangential parts of the teeth are drawn to the centre.

With involute teeth, as before stated, the path of contact of the teeth is a straight line, giving a constant angle of obliquity of action. Herein they differ from cycloidal teeth, which have a variable angle of obliquity, their greatest obliquity of action being at the beginning and end of contact of the teeth, while as they pass the point of contact of the pitch circles the angle is reduced to nothing, and their line of action becomes tangential to those circles. Obliquity of action is an objectionable feature in wheel gearing, for at the acting surfaces of the teeth it causes a component of the total pressure transmitted to act, so as to increase the pressure between the axle of the wheel and its bearings.

All involute teeth of the same pitch work correctly together. An important property possessed by involute teeth, and by no other form, is, that the distance between the centres of the wheels may be either slightly diminished or increased without altering the smoothness of action of the teeth, or the uniformity of their angular velocity.

When involute teeth gear with a rack, the teeth of the rack are drawn by straight lines, which make the same angle with the vertical as the angle of obliquity of the wheel. (See Worm and Wheel, Plate XLVI.)

Mortise Wheels.—In the previous examples the teeth have been assumed to be cast solid with the rim of the wheel. But where wheel gearing runs at high velocities, wooden teeth are made on one wheel to gear with iron teeth on its fellow.

The wooden teeth are usually made of ***hornbeam***, and are let into grooves cast in the iron rim of the wheel to receive them. The iron rim of a mortise wheel is made deeper than usual to provide a firm bed for the tooth, and the wooden tooth is made thicker than its corresponding iron tooth in the proportion shown on the sketch, no clearance being left between the teeth.

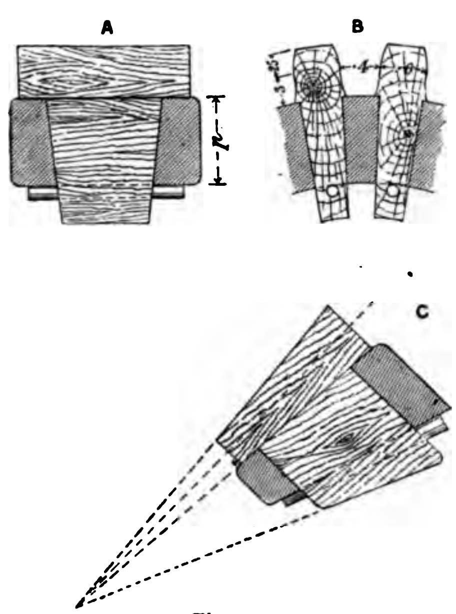


Fig. 94.

The method of securing teeth by a pin at the back of the rim is a common and effective one. Fig. 94, A and B, are two views of the teeth for a mortise spur wheel; and C is the tooth of a mortise bevel wheel.

49. HELICAL GEARING. PLATE XLV.

Screw Gearing, with parallel axes, similar to that illustrated in the Plate, is now coming into very general use. In 1666 Dr. Hooke invented the stepped tooth wheel, in which the teeth, instead of being placed on the rim parallel to the axis, are made in the form of a succession of advancing steps in the direction of the circumference of the wheel (Fig. 95A). This wheel resembles a series of exactly similar toothed discs, placed side by side on the same axis, the teeth of each disc being successively placed a little in advance of the preceding. If now this stepped wheel be made to gear with a similarly constructed wheel, the parts of the steps in advance will engage first, the remaining steps coming successively into gear, thus combining the smoothness of action of small teeth with the strength of large teeth. If the number of steps (or assumed discs) be infinitely numerous, the whole of the steps in immediate contact will form one continuous tooth across the rim of the wheel in the form of a helical curve (Fig. 95B); and a wheel with teeth of this kind is really a short piece of a many-threaded screwed cylinder, the threads having a very large pitch (Fig. 96). Such a wheel having teeth forming right-handed threads will gear with another wheel having left-handed threads, provided both wheels have the same obliquity of tooth; and the velocity ratio of the wheels will, as with ordinary gearing, be inversely as their radii. The contact of each pair of teeth commences at the forward edge of the helix, and terminates at the hinder edge. The objection to these wheels is that a component of the total force transmitted through them is expended in causing an end thrust.

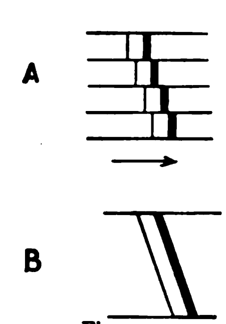


Fig. 95.

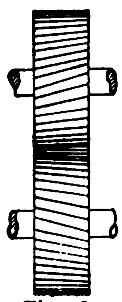


Fig. 96.

This objection is overcome by an arrangement which is equivalent to combining side by side with the first pair another pair of helical wheels with teeth of opposite obliquity, one side being a right-handed and the other a left-handed screw thread, forming a V-shaped double-helical tooth on the rim of the wheel, as shown in Plate XLV.

The form of these teeth is exceedingly strong, and when correctly constructed their action is very smooth and noiseless.

TO DRAW THE EXAMPLE (Plate XLV.), commence with the elevation of the pair of wheels in gear, setting off the teeth on the pitch line, as in any ordinary case, and using preferably the involute form of tooth (see page 120).

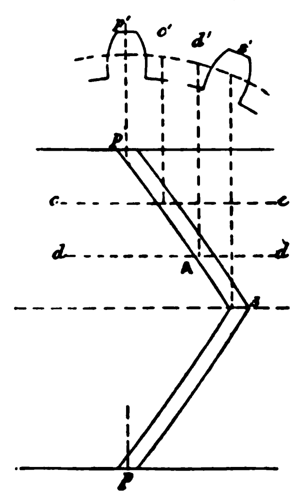


Fig. 97.

Consider the tooth A (Fig. 97), which forms part of a double screw thread on the face of the wheel. It commences at p in the plan, and in crossing the rim advances to s (see "Screw Threads," page 35), and the elevations of these two positions are shown at p', s'. If this part of the tooth be cut by any number of planes, $c\ c$, $d\ d$, at right angles to the axis, the projection of the section of the tooth on the elevation would be in some position $c'\ d'$ between p' and s'. Reversing this way of considering the tooth, if p' and s' in the elevation be given, we can obtain p and s in plan, and by dividing the part of the pitch circle between p' and s' in the elevation

into any number, say three equal parts, and the half face of the wheel in plan into the *same* number of equal parts, by planes *c c*, *d d*, we get intermediate positions of the tooth in both elevation and plan

In the example given in Plate XLV., the width of the rim, Fig. 98, is four times the pitch, and the advance of the tooth *a*, measured on the pitch line = 1·4 times the pitch, giving an angle between the tooth and edge of the rim of 55°. This angle has been recommended by Mr. Springer as a suitable constant angle for all pitches. (See *Mechanical World*, vol. iii. No. 26.) The width of the rim may be increased for very heavy work.

Referring to the Plate (XLV.) On the elevation of the pitch line of the wheel from *a'*, draw a tangent *a b* = 1·4 of the pitch, and measure this distance along the circumference, namely, *a' c'*. Divide *a' c'* into, say, three equal parts. Divide the distance *a c* in the plan into the same number of equal parts; then the elevation of the teeth already drawn will give points on *a a*, *a a* in plan. Now trace the elevation of the teeth on tracing paper, and move the paper about the axis of the wheel till *a'* coincides with *c'*, and project from the tracing paper for points of the teeth on *c c* in plan. Similarly intermediate points on *d d*, *e e* in plan may be obtained by moving *a'* to *d'* or to *e'*. The points may be joined by a French curve.

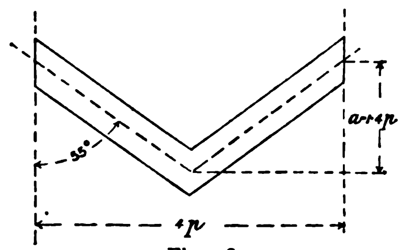


Fig. 98.

50. WORM AND WHEEL. PLATE XLVI.

Worm gearing consists of a screwed bar called a ***worm***, which is used to drive a wheel, the teeth of the wheel being so formed as to suitably engage the thread of the worm. The arrangement is used for obtaining large mechanical advantage, by converting a high into a low angular velocity. One complete rotation of the worm will cause the wheel to rotate through an arc measured on its pitch line equal to the pitch of the thread of the worm. A single-threaded worm may be considered to be a wheel with one tooth; a double-threaded worm (as in the example, Plate XLVI.) a wheel with two teeth, and so on. Thus if a single-threaded worm gear with a wheel of 30 teeth, then for one revolution of the worm the wheel makes $\frac{1}{30}$ of a revolution; that is, a velocity ratio between wheel and worm of 30.

If the worm were double-threaded, the wheel would make $\frac{1}{15}$ of a revolution; and the velocity ratio would be 15.

The ***axial pitch*** of the screw thread of the worm is the pitch of the helix measured parallel to the axis. When the screw has more than one thread, the axial pitch divided by the number of threads (namely, two if double-threaded, or three if treble-threaded) is called the ***divided axial pitch***, and this pitch is equal to the ***circular pitch*** of the teeth of the wheel; that is, the circumference of the pitch circle of wheel divided by the number of teeth. In worm gearing the worm drives the wheel; but unless the pitch is very coarse the wheel will not drive the worm. This property is frequently of great value, as the worm locks the wheel in any required position, and owing to the great difference between the angular velocity of the worm and wheel, the movement of the wheel can be regulated with any required degree of minuteness.

The movement of the wheel is evidently due to the sliding contact of the worm thread against the wheel teeth, hence the friction and wear and tear are very great. The worm is usually steel, and the cast-iron wheel is fitted with a brass-toothed rim.

The teeth of worm wheels are frequently made of the same section throughout, the wheel being similar to an ordinary toothed wheel, except that the teeth are inclined, so as to coincide with the angle of the worm thread. In such a case the wearing surface is very limited, the contact being confined to the centre of the wheel teeth. In order to increase the length of contact or wearing surface, the worm is, so to speak, inserted further into the rim of the wheel, and for this purpose the rim and wheel teeth are made concave on their outer surface by arcs struck from the worm centre, as shown in the left-hand view on the Plate. By this means the wheel teeth are made to engage the worm thread along their whole length, and the form of the teeth, when correctly made, will now be found to vary from end to end.

Professor Willis first pointed out that if a section be taken through the axis of the worm, perpendicular to the axis of the wheel, and the teeth of the worm and wheel be made like those of an ordinary rack and spur wheel, the two will gear correctly.

This is correct so far as it goes, namely, for the section on the plane through the axis of the worm; but the correct method of obtaining the true form of the wheel teeth on sections perpendicular to the wheel axis other than that through the axis of the worm—a point of considerable importance, which seems to have been overlooked by writers on the subject—was first explained by Professor Unwin, in his work on "Machine Design," page 298, where he points out that "*all* the sections of the worm threads and wheel teeth, on planes normal to the wheel axis, must be of forms suitable for a spur wheel and rack of the same pitch."

To draw the Worm and Wheel.—The example illustrated in the Plate is given for the purpose of explaining the method of drawing the worm and wheel correctly. The student may, with advantage, use a larger wheel than the one chosen (owing to limited space), namely, one having say thirty teeth instead of twenty. The length of the worm is also greater than is necessary in practice, it being sufficiently long when it exceeds the limits of the path of contact of the wheel teeth. The reason for extending it is merely for convenience of drawing, as will be explained below. First draw the centre lines of the worm and wheel, and set off lines representing the pitch lines, and the tops and bottoms of threads of the worm, to the dimensions given; also the pitch circle P P of the wheel. Let the perpendicular C *o* through the centre of the wheel cut the pitch line of the worm in the point *o*, and through this point draw a line inclined 15° to the vertical, or 75° to the horizontal. This line will represent on a plane through the axis of the worm one face of the tooth of the worm (which is here treated as a rack), suitable to gear with an involute tooth on the wheel (see page 122). Mark off from the point *o* the thickness of tooth on the pitch line, namely, 0·48 of the (divided axial) pitch $= 0{\cdot}48 \times 1{\cdot}25 = 0{\cdot}6$ inch; finish this section of the tooth, and repeat it throughout. (Fig. 99.) Complete the helices as shown by the method explained for square-threaded screws, Plate VII. The screw in the present example is double-threaded.

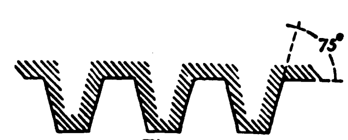


Fig. 99.

Take sections through the worm and wheel (Plate XLVI., Fig. 1) by planes ***m m***, ***n n***, ***o o***, ***p p***, ***r r***, parallel to the plane of the wheel. The object of this is to find the true form of the section through the worm thread at various positions along the length of the wheel teeth. Take for example the section ***m m*** (Fig. 1). This

section cuts the worm thread at points *a*, *b* and *c*, namely, at the root of the thread in *a*, on the pitch line in *b*, and on the outer circle at *c;* and to obtain a side elevation of the section (M., Fig. 2), project from these points to the corresponding points on the helices, namely, from *a* (Fig. 1), to the root helices (Fig. 2), from *b* to the pitch line helices, and from *c* to outer helices of the worm tooth. All the other worm sections (N O P R, Fig. 2) are obtained in the same way. By the aid of these sections we can now obtain the form of the wheel tooth suitable to gear with the worm section at the corresponding positions in each case, as a spur wheel gears with a rack.

Now the action of the worm upon the wheel is exactly the same as though the wheel were driven by an endless rack; hence the relative motion of wheel and worm is unchanged, whether the wheel rotates on a fixed axis and the rack moves longitudinally (as it practically does with the worm), or the worm remains stationary while the wheel rolls along on the worm.

Let now the pitch circle of the wheel roll along the pitch line of the worm. In order to do this, mark off on each side of *o* on the worm pitch line equal distances, 1 2 3 4, &c., 1′ 2′ 3′ 4′, &c., and similar equal divisions on the horizontal centre line of the wheel. Now take a piece of tracing paper, draw on it the pitch circle of the wheel, and mark equal divisions on the tracing paper circle, so that as it rolls along the worm pitch line the respective divisions may coincide. The centre of the pitch circle on the tracing paper will successively occupy the positions 1 2 3 4, &c., on the horizontal line through the centre of the wheel. Commence with, say, section M. Place the tracing paper so that the divisions on the tracing paper pitch circle successively coincide

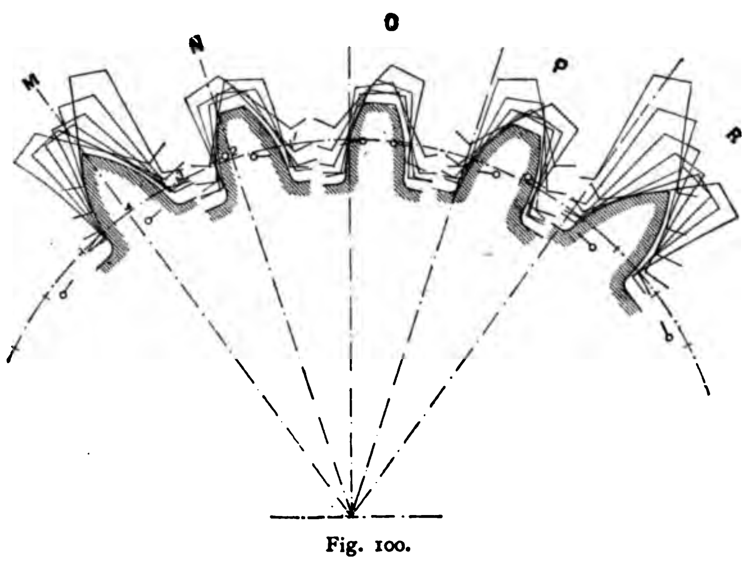


Fig. 100.

with the respective divisions on the worm pitch line, and trace the form of the section M for each of the successive positions of the tracing paper. These will appear as shown at M, Fig. 100, and they represent the successive positions of the section M of the fixed worm relative to the rolling worm wheel, and the contour of the tooth of the wheel may now be drawn to *touch* the figures on one side, clearance being left on the other as for wheel teeth. The same process is now repeated

for the other sections, and we obtain the shape of the wheel tooth in five different positions throughout its length. Templets may be made of the forms thus obtained with which to make the patterns of the teeth. The forms of the sections at M and R are given for the teeth as though they were not cut off taper at the sides, as shown at Fig. 1. They may afterwards be cut as desired, without interfering with their correct action.

To draw the elevation of the wheel teeth as shown in the lower part of the Plate: let the centre lines O M, O N, O *o*, &c., on the tracing paper, Fig. 100, be made successively to coincide with C B, the centre line of the wheel, Fig. 2, and mark or prick through the tracing paper the sections of the teeth one upon the other. From these combined sections the side view of the teeth may be obtained. The outer extremities of the teeth are tapered off, as shown on Fig. 1, from which view the addendum or outer circle can be obtained. The teeth on the lower part of Fig. 1 are projected from Fig. 2.

Chapter IX.

51. THE LATHE. FIG. 1 (FRONTISPIECE), PLATES XLVI. TO LI.

In the early days of engine construction everything depended upon the manual skill of the workman, and the tools and appliances which were then available were of the roughest kind. But as increased care began to be bestowed on the construction of machine tools, so the quality of manufactured machinery improved, until a degree of accuracy, precision, and finish has now been attained which was undreamt of in the days of Watt and the early engineers. This advancement towards perfection is attributable to nothing so much as to the improved construction of that most important of all the machine tools, the *lathe*. With the introduction of the *self-acting slide rest* began a new era in construction, for by it an absolutely accurate mechanical method of holding and guiding the cutting tool was now substituted for the merely approximate accuracy of the hand tool.

The essentials of a lathe are:—A pair of headstocks, a rest, a bed on which the headstocks and rest are placed, and legs or standards to support the bed. The axis or principal centre of the lathe coincides with the axis of the spindles in both the fast and loose headstocks. This centre line must of course be parallel to the surface of the bed, and to the edges of the bed which guide the slide rest. The size of a lathe is described by the height of the centre from the top of the bed. The work to be turned is suspended between the two headstocks, so that the axis of the figure to be turned coincides with the centre line of the lathe. This is arranged by marking the centre of the work at each end beforehand, making two small holes at these centres, and placing them between the centres which protrude from each headstock, as shown at *c*, Plates XLVIII., XLIX. and Frontispiece.

The work is made to rotate by means of a belt, which is driven independently of the lathe, and which passes over the speed cone, and gives motion to the spindle of the fast headstock. The rotation of the spindle is transferred to the work by means of a driver, or pin, passing through a hole in the face plate, and coming in contact with a carrier fastened to the object to be turned. The tool is held firmly in the slide rest, and when everything is ready, and the lathe set in motion, the tool is now placed against the work. If the cutting edge of the tool be advanced towards the axis of the lathe (termed *feed or surfacing*) the work will be cut in circular peripheries of continually decreasing diameter. If the cutting edge of the tool be advanced in the direction parallel to the axis (termed *traversing*), the tool cuts a perfectly cylindrical surface. By a combination of these two motions bodies may be turned *taper* or *curved* as required.

52. THE LATHE BED. PLATE XLVII.

53. THE FAST HEADSTOCK. Plate XLVIII.

This headstock is placed at the left-hand end of the bed, and is secured in this position, so that the axis of the spindle S is perfectly true and parallel with the side and top of the bed.

The *spindle* or *mandril* LS is in the main centre line of the lathe, and is supported at each end of the headstock in bearings. At the extreme right of the spindle it is screwed to receive a chuck or face plate FP, which is screwed up against the collar at the back of the thread. Projecting from the front of the spindle is the pointed centre piece C, which is made conical as shown, and driven into the nose of the spindle, bored taper to receive it. At the right-hand end, and behind the collar, the spindle is turned taper, and works in a corresponding taper in the brass bush. The bush is a solid ring, turned taper inside and outside, and is driven into the casting, which is bored out to receive it. At the left-hand bearing the spindle is fitted with a steel cone, which slides loosely on the spindle, but is prevented from turning round on it by a key. The cone is made taper in the opposite direction to that of the right-hand bearing. Wear is taken up by bringing these two cones closer together, and in order to do this the spindle is screwed on the left of the left-hand bearing, and two shallow nuts are provided for tightening up the cones to their bearings. The spindle itself ends just beyond the wheel, as shown by the dimension line giving the extreme length of the spindle. Here it bears against the flat end of a short steel pin. This bearing pin has its ends screwed, and is secured by two nuts to a metal bridge, which is held in position by two little pillars (seen best in the plan and in Fig. 1). The pillars are screwed into the casting. The end thrust on the spindle, due to the action of the cutting tool, is carried partly on the right-hand conical bearing, but chiefly by the steel pin at the left-hand extremity of the spindle, which can be adjusted by means of the nuts, so that the spindle shall not bear excessively on the right-hand conical bearing.

Immediately to the left of the right-hand bearing is a spur wheel W′ keyed firmly to the spindle. Adjoining it is a speed cone pulley SC, having three steps to regulate the speed, and solid with the pulley is a toothed pinion P, shown clearly on the plan. The speed cone and pinion run loose on the spindle. The speed cone may be attached to the spur wheel W′ by means of a bolt B, which slides in a groove in the wheel, shown in the sectional view, also in the left-hand bottom corner of the drawing. The bolt is here shown dropped down in the groove, and the spur wheel and cone pulley are therefore "out of gear," or disconnected. On the front end of the cone pulley is a rim, and in the rim two slots or openings. When it is desired to connect the wheel W′ and the cone pulley, the bolt B is slipped into one of these slots and the nut is tightened up. They are then connected.

If the speed could be varied sufficiently by means of the three steps on the cone pulley, there would be no need for the spur wheels and pinions, and the pulley could be keyed to the spindle at once. But frequently a much slower motion of the spindle is required, and to obtain it *back gear* is used. This consists of a back shaft, to which are secured a wheel W, and pinion P^1, arranged to gear with the wheel W^1 and pinion P on the spindle. The back shaft is made eccentric in its bearings (see end view), and by means of a handle the toothed wheels can be thrown in or out of gear. A pin P passing through the shaft, secures it in one of two positions, namely, either in or out of gear. Suppose everything disconnected, the back gear thrown out, and the pulley and wheel

uncoupled. Then, if the belting were in motion, the cone pulley and pinion would rotate loose on the spindle without setting any other part of the lathe in motion. If the wheel W^1 be now coupled to the pulley, the lathe spindle itself will rotate at the same speed as the pulley, and the more quickly the smaller the step on which the belt runs.

If now the wheel W^1 be again disconnected from the pulley, and the back shaft and wheels be thrown in gear, the motion of the spindle will then be that derived from the circuit of gearing, P and W, and P^1 and W^1. The cone pulley drives the back shaft slower than itself in the ratio of P to W, or, in the example chosen, $\frac{16}{48} = \frac{1}{3}$; and P^1 (being keyed on the same shaft as W) drives W^1 in the ratio of P^1 to W^1, namely, $\frac{16}{48} = \frac{1}{3}$ as before. But W^1 is keyed on the spindle, and the spindle therefore rotates slower than the cone pulley in the ratio $\frac{1}{3} \times \frac{1}{3} = \frac{1}{9}$.

The two set screws, shown underneath and at each end of the headstock, are to adjust its centre line with the centre line of the bed.

The wheel gearing shown at the extreme left of the lathe conveys the motion of the main spindle to the leading screw L S, and causes the rotations of the leading screw to bear a certain proportion to the rotations of the lathe spindle. It further regulates the rotations of the back shaft B S, and thereby also the rack traversing and surfacing feed motions, explained below.

Screw Cutting.—Suppose a cylindrical bar, upon which it is required to cut a screw thread, to be placed in the lathe between the centres. If the bar rotate and the tool be pressed against it without being caused to move at the same time in the direction of the axis of the bar, it will cut a circular groove upon the bar. If the lathe cease to rotate, and the tool be pressed against the work, and at the same time be made to move in the direction of the axis of the bar, the tool will scratch a straight mark on the surface of the bar parallel to the axis.

If, now, these two motions be combined, namely, the rotation of the bar and the sliding motion of the tool, a *helical or screw thread* will be cut on the surface of the bar; and this thread will be fine or coarse in pitch according to the distance travelled by the tool during a revolution of the bar. But the sliding movement of the tool is obtained by the rotation of the leading screw, which works in a nut attached to the slide rest; hence the pitch of the thread cut by the tool upon the bar will depend upon the ratio between the rotations of the lathe spindle and the rotations of the leading screw; and this ratio depends upon the number of teeth on the wheels E and B (Fig. 1), or on the combination of wheels by which these may be connected. If the pitch of the screw required be the same as that of the leading screw, then for one revolution of the spindle there must be one revolution of the leading screw, and this is obtained by placing a wheel B on the end of the leading screw, having the same number of teeth as E (the revolutions of which are the same as those of the spindle), and connecting the two by an "idle" wheel C (see page 106). If the pitch required is to be half that of the leading screw, then the tool must advance only half the distance previously advanced, that is, the leading screw must rotate with only half the velocity of the spindle, and a wheel will be substituted at B having twice the number of teeth in wheel E; so whatever the ratio between the pitch required and that of the leading screw, the same ratio must be made to exist between the rate of revolution of the leading screw and that of the lathe spindle. The intermediate gearing, or train of wheels required to connect E and B, is mounted on the "radial arm," which contains slots for moving the wheels of various sizes into any required position. The "radial arm" swings round the end of the leading screw as a centre, and is itself secured by two bolts, as shown on Fig. 1. and on Plate XLVII.

When a *left-handed* thread is required, the leading screw must rotate in the opposite direction to that of the spindle, so that the tool will then move along the bed from left to right, instead of in the opposite direction. The method of accomplishing this reversing motion of the leading screw will be understood by reference to the four small equal wheels immediately to the left of the headstock (Fig. 1), and shown at K F G E on Plate. The two middle wheels of the four are "idle" wheels, geared together, and attached to a small swinging plate, which swings about the centre of wheel E. This plate can be so placed that wheel E shall receive its motion from K through one of these middle wheels only—in which case E and K will rotate in the *same* direction; or through the two wheels, as shown in the engraving (Fig. 1), in which case E and K will rotate in *opposite* directions, as required for cutting left-handed screws.

54. THE LOOSE HEADSTOCK. Plate XLIX.

The loose headstock, or poppet head, may be moved along the bed, so that pieces of work of different lengths may be held between the centres. It is secured to the bed in any required position by means of the long bolt in the centre of the foot, which clamps it to a cross-piece spanning the under-side of the bed, as shown on the end elevation. This longitudinal movement of the headstock is sufficient for rough adjustment, but it is further supplemented by a movement of the centre *c*, which is made taper, and fitted in the end of the barrel B, the barrel being made to slide in or out of the headstock by means of the long screw turned by the wheel shown to the right. The right-hand end of the barrel is screwed, and forms the nut for the screw. On turning the wheel the screw itself cannot move longitudinally, owing to a collar on the screw on the one side, and the boss of the wheel keyed to it on the other side of the plate *d;* but the barrel, which acts as the nut, moves in or out as required. The barrel is prevented from turning round with the screw by means of a longitudinal groove or slot cut along the bottom of the barrel. This groove fits a T-headed key *k*, which is secured to the casting below the barrel. An arrangement for clamping the barrel in any desired position is provided, consisting of an open slit on one side in the left-hand end of the casting which encircles the barrel. On each side of the slit a lug is cast. A screwed bolt, terminating in a handle *h*, springs the lugs together and clamps the barrel. It will be noticed that the foot of this headstock is made in two parts, so that the upper part may slide on the lower. The object of this is to provide a means of taper turning by causing the back centre to be out of truth with the main centre of the lathe. If the back centre is drawn towards the front of the lathe by means of the small bolt *m*, shown in the side of the foot of the headstock, the ordinary parallel traverse of the tool will evidently turn a smaller diameter at the back end than at the front end of the work. It is of great importance in parallel turning that the two centres in the fast and loose headstocks should coincide with the main centre of the lathe, which should of course be parallel to the edges of the bed.

55. THE SLIDE REST. Plates L. and LI.

The object of the slide rest is to hold the cutting tool securely, and to cause the tool to move with mechanical precision in a direction either parallel to the axis of the lathe, or at right angles to it. It consists of a series of plates placed one upon the other. The top plate *m* contains two

square-headed bolts, which are screwed down on the cutting tool, and secure it. This plate may be turned through any angle about the large centre bolt *n* (Plate L.), so as to secure the tool in any required horizontal position. Next follows the upper slide *p*, which carries the tool and the plate *m*.

The upper slide, which is guided by dovetailed grooved surfaces, is made to move to the right or left, parallel to the axis of the lathe, by turning the handle, seen to the right of the slide rest. The handle is fixed on the end of a long screw, so that handle and screw turn together, thereby causing the nut, which is formed at the lower end of the large bolt *n* and which clasps the screw, to slide to or fro along the screw, and to carry the whole upper rest with it.

An arrangement is made for giving the tool, when required, an oblique feed motion relatively to the axis of the lathe. This is done by turning the upper guide plate *r* round a vertical pin (shown screwed into the centre of plate *w*), and securing it in any position by means of two bolts, which may be moved round the centre in a circular groove *g*.

56. THE SADDLE. Plate LII.

Below the upper slide and guide *p* and *r* (Plate L.), is another sliding pair, *w* and *s*, acting perpendicularly to the upper pair, and guided by similar dovetailed surfaces. The lower guide *s* forms the saddle of the lathe, and is seen in Fig, 1, or in Plate LII., where the upper portions of the rest have been removed. The long screw, which runs the whole length of the saddle (Plate LII.), engages with the nut *y*, shown below plate *w* (Plate L.) This nut is formed at the end of a wrought-iron pin driven in on the under side of plate *w*. By turning the handle at the front end of the screw, the slide rest is moved in a direction at right angles to the axis of the lathe. The saddle is fitted to slide on the bed itself, being dovetailed on the front, and having a bevelled packing strip the whole length of the saddle at the back.

Self-acting Motion of the Slide Rest.—It has already been shown how the slide rest may be moved *by hand*, either parallel to, or at right angles to, the axis of the lathe. But when the lathe is so fitted that the motion of the slide rest can be obtained and regulated by mechanism within the lathe itself, it is said to be *self-acting*.

Motion of the slide rest perpendicular to the axis is called *surfacing*; motion parallel to the axis is termed *sliding* or *traversing*. The surfacing motion of the tool is made self-acting as follows :—

On the back end of the long screw *Z* in the saddle (Plate LII.) a small pinion *e* runs loose, motion being communicated to it through the gearing on the back shaft BS. When the motion of the slide rest is required to be self-acting, this pinion is secured to its spindle (which is a continuation of the screw) by tightening up the winged nut *f*. This brings the pinion and the collar on the end of the screwed spindle into close contact, thus setting up friction sufficient to cause the motion of the wheels to be transmitted to the screw, and through it to the slide rest. By this means the surfacing motion of the rest can be made self-acting. To prevent the saddle from slipping back when the tool is pressed against the work, the three small set screws (shown at the back of the saddle) are tightened up against the packing strip, and the saddle is thereby gripped to the bed.

The self-acting *traversing* motion of the rest is obtained in either of two ways:—At the front of the lathe it will be seen that the bed is provided with a leading screw L S, and a rack R.

The leading screw is used for self-acting traversing for cutting the threads of screws only. The rack provides another means of self-acting traversing for general purposes, and thus enables the leading screw to be kept in good condition.

To traverse with the leading screw, we must have some means of connecting or disconnecting the slide rest with that screw at once if required. This is done by clasping the screw between the two halves of a horizontally divided brass nut, as shown on Plate LI.

The handle L H is keyed to a short spindle, with a circular disc at the end from which two pins project. These pins fit into grooves made in each half-nut to receive them. In the Plate LI. the half-nuts are shown raised, and the slide rest is therefore disengaged from the screw. On pressing the handle down, the nuts engage the thread, and the slide rest will now be subject to the action of the leading screw, moving parallel to the bed with a velocity depending upon the speed of rotation of the screw; and, as before explained, this may be regulated by change wheels connecting the ends of the lathe spindle and leading screw.

To Traverse with the Rack.—This may be done *by hand* by turning the long winch-handle W H (Plate LII.) A pinion at the other end of the spindle, which carries this handle (see sectional view in Plate), is in gear with the rack R, and any movement of this pinion causes a sliding motion of the whole carriage along the bed of the lathe in either direction. This motion may be rendered self-acting by connecting in some way the motion of the lathe mandril with the spindle which carries the pinion *n* in gear with the rack (Plate LII.) This is done by first securing the large toothed wheel W_2 to this spindle. In the drawing (Plate LII.) the wheel is shown loose on the spindle. To secure it, the winged nut *m*, when screwed up, tightens it on the cone B (an enlarged view of which is given separately). The connection required between the lathe mandril and the rack is thereby completed, for the motion of the lathe mandril being communicated by the wheel (Fig. 1) on the back shaft B S, is transferred through the worm *k* and the wheels *h* and *g* (shown dotted on back view of saddle in right-hand bottom corner of Plate LII.) But wheel *g* is on a spindle which runs through the saddle, and has a pinion A at its other extremity, and this pinion gears with W_2, and thereby transfers the motion to the pinion *n*, in gear with the rack. The worm *k*, which is driven by the back shaft, is required to slide along this shaft with the saddle as it glides along the bed. This is accomplished by cutting a long key-way in the back shaft in which the pinion key may slide.

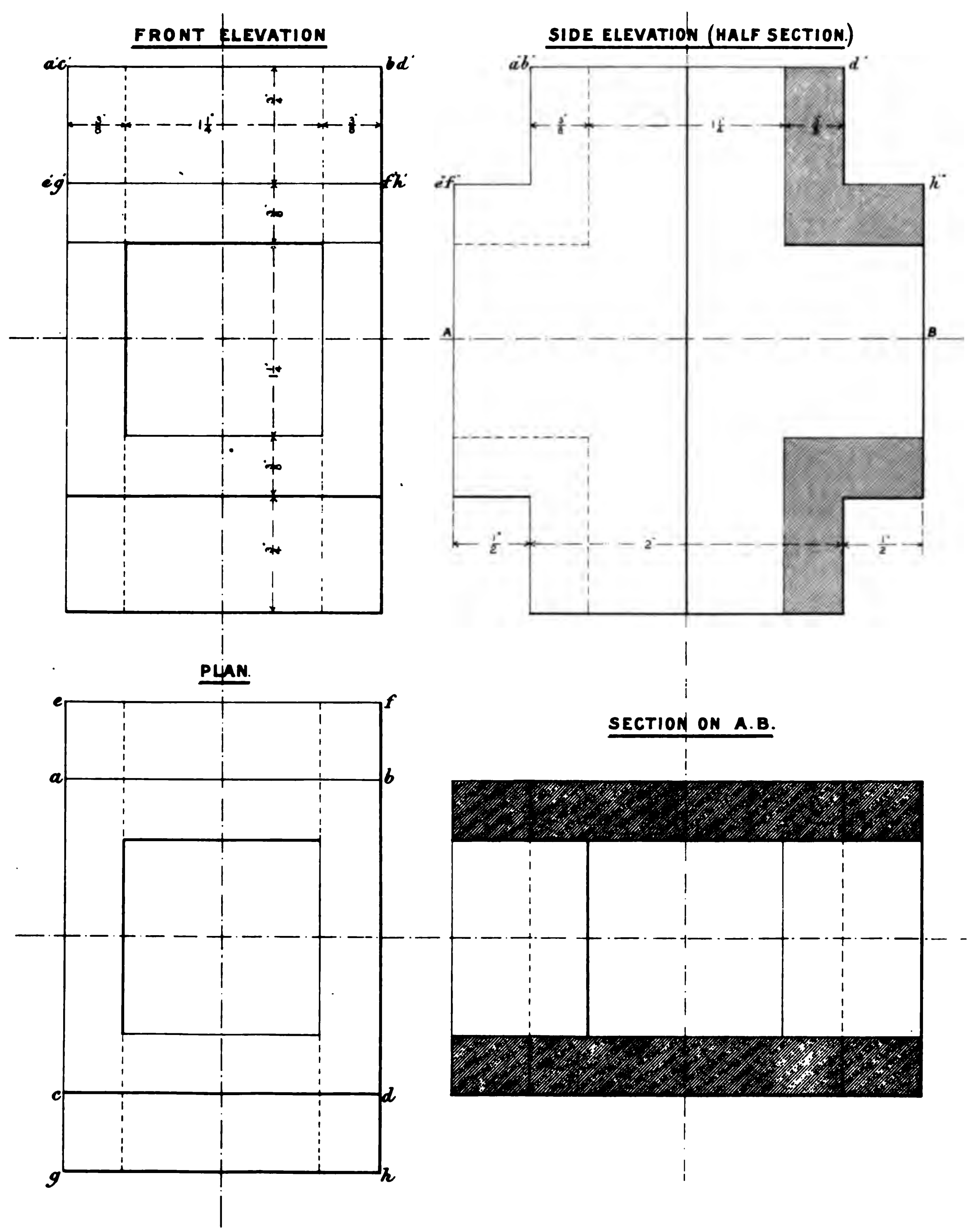
PAWSON AND BRAILSFORD, PRINTERS, SHEFFIELD

INDEX.

INDEX.

PROJECTION.

PLATE I

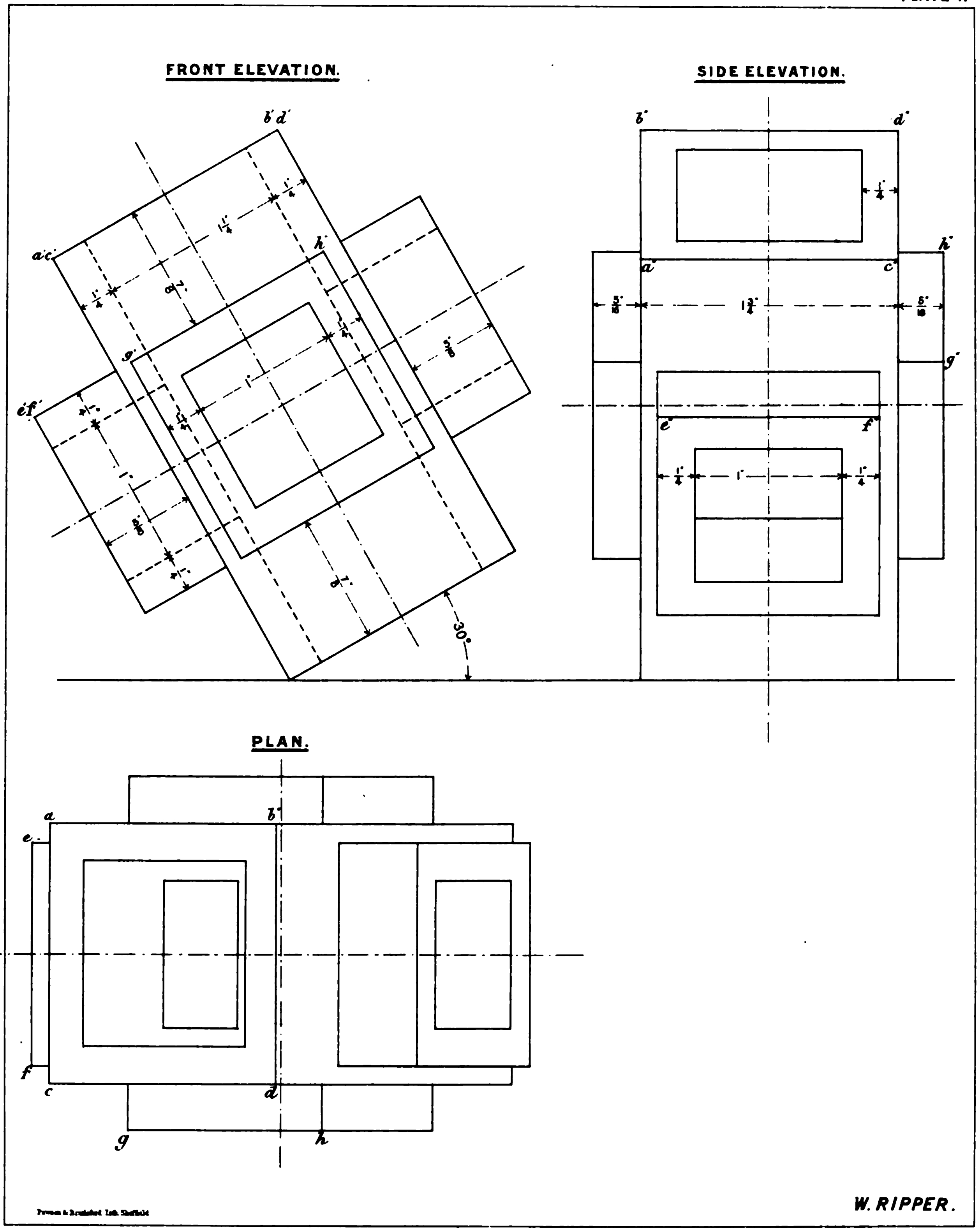


W. RIPPER.

Draw full size. Work to dimensions.

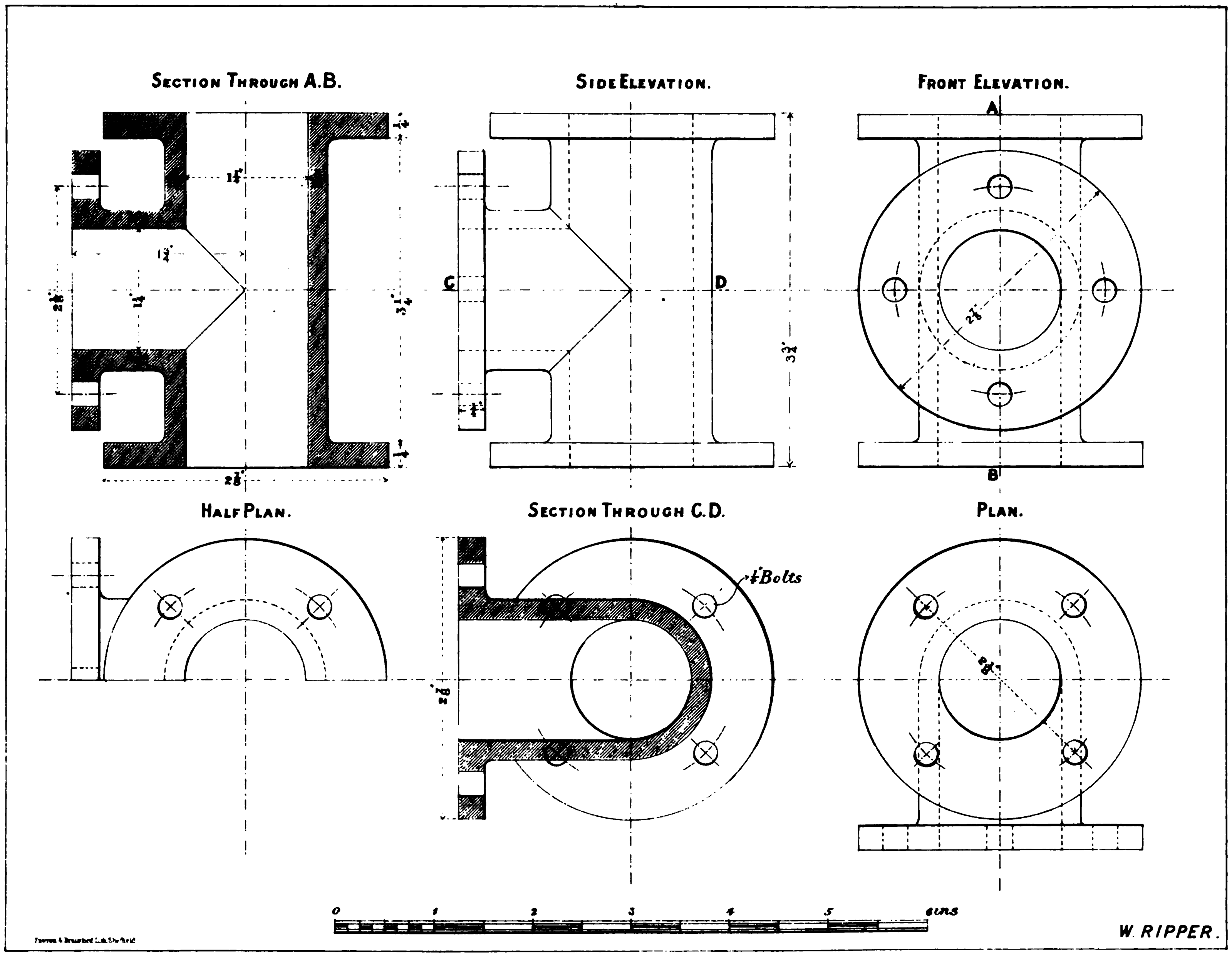
PROJECTION.

PLATE II



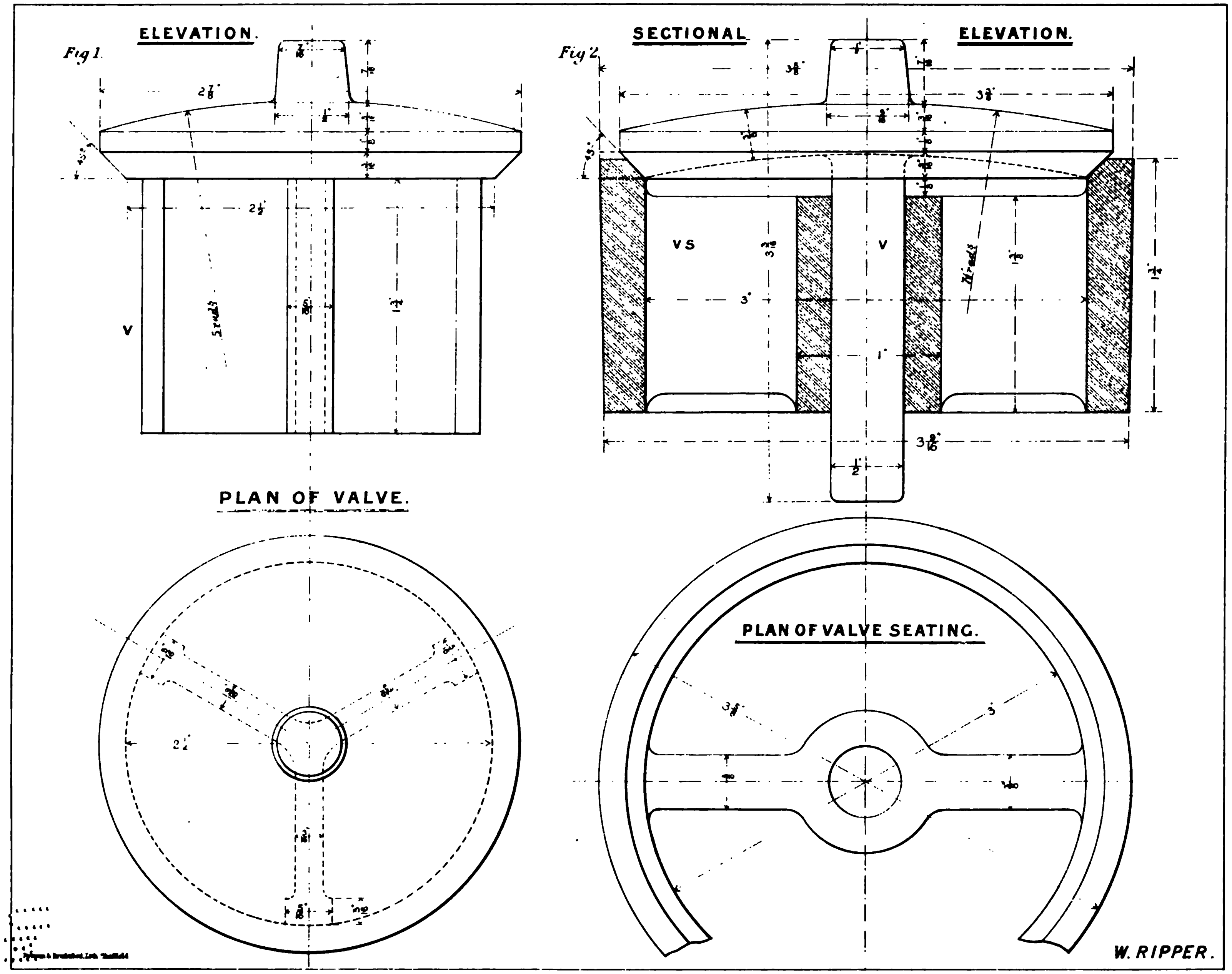
Draw full size. Work to dimensions, and project one view from another.

PROJECTIONS OF A T JOINT.



Draw full size. Project the views one from the other.

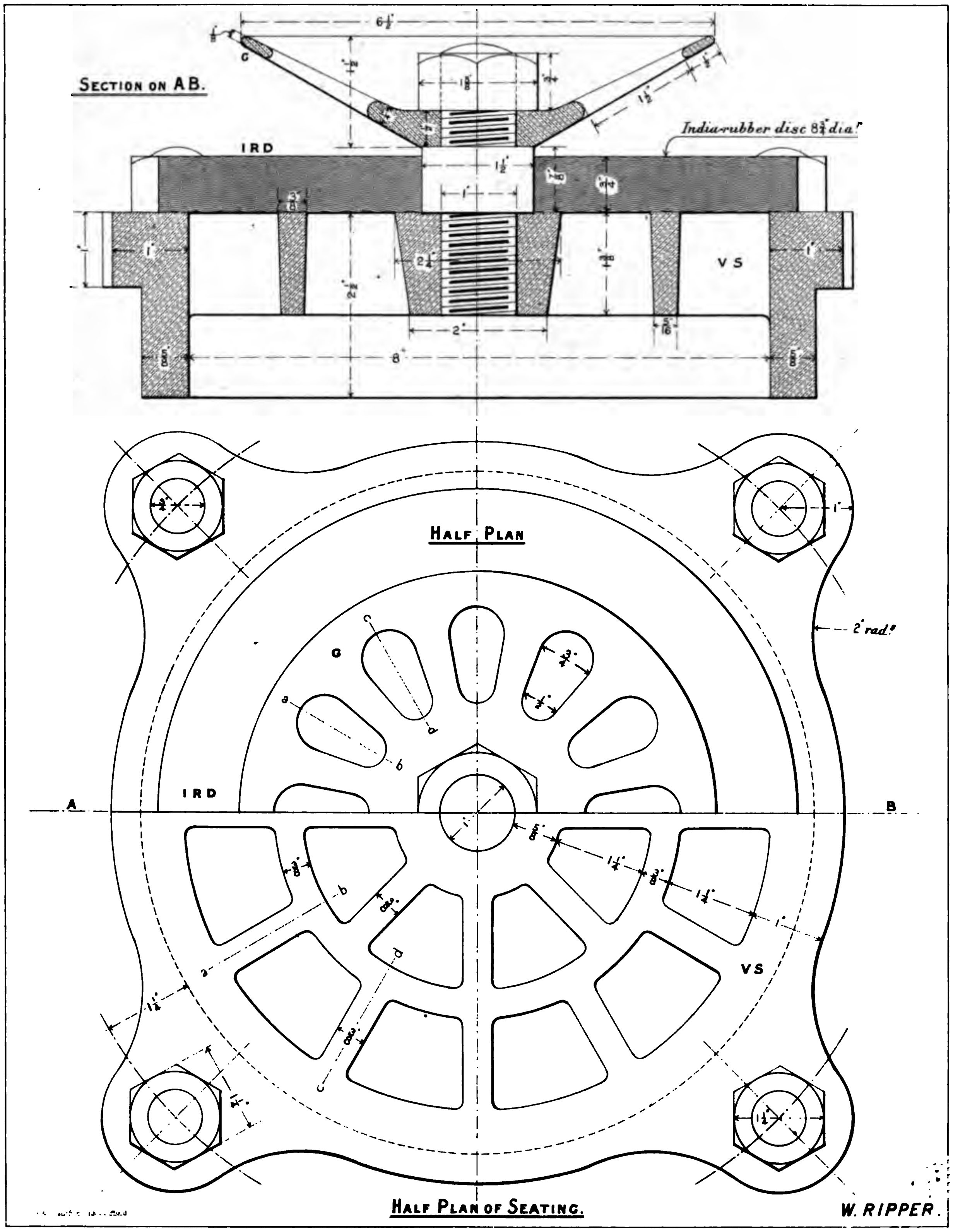
Conical Disc Valves.



V valve. *VS valve seating.* *Draw full size.*

INDIA-RUBBER DISC VALVE.

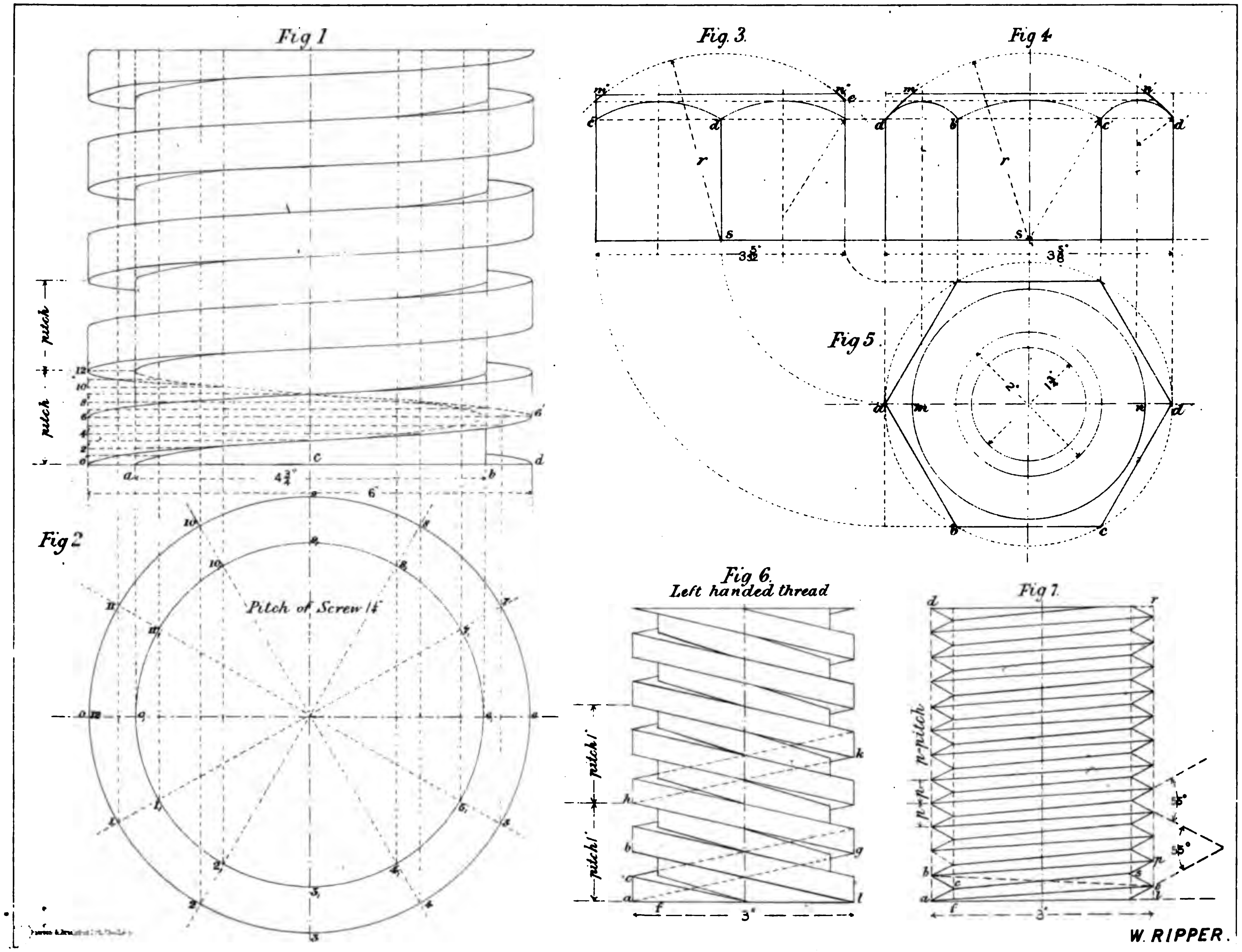
PLATE V



V S valve seating. G guard. I R D india-rubber disc. Draw full size.
For proportions of hexagon nuts see Plate VIII. and Table IX. a b makes 30° and c d 60° with A B.

SCREWS AND NUT.

PLATE VII



Figs. 1 and 2.—Elevation and plan of square-threaded, right-handed screw. Figs. 3, 4, 5.—Hexagon nut.
Fig. 6.—Double-threaded, left-handed, square-threaded screw. Fig. 7.—V-threaded screw, 3½ threads to the inch. Draw full size.

BOLTS AND SCREW THREADS.

PLATE VIII

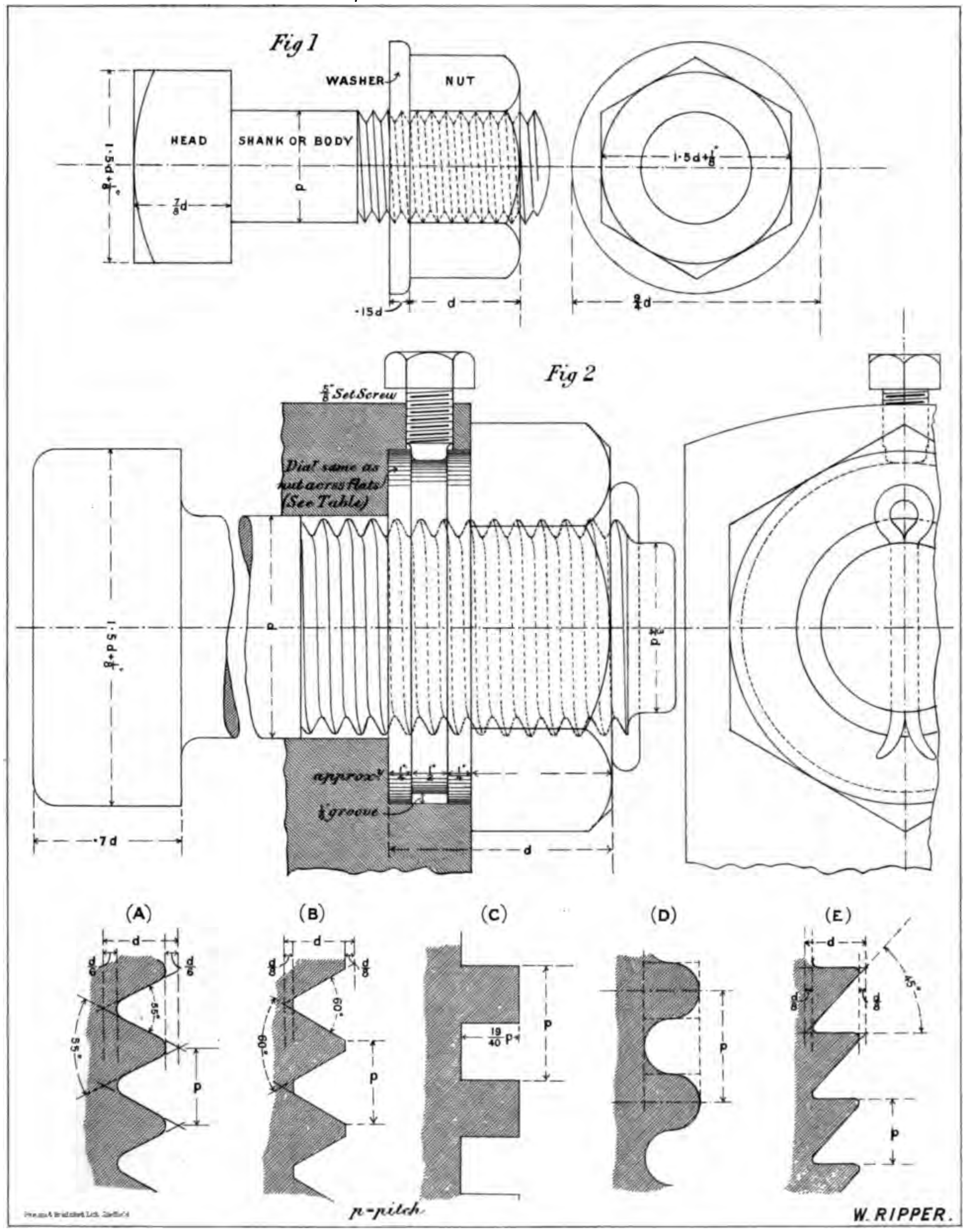
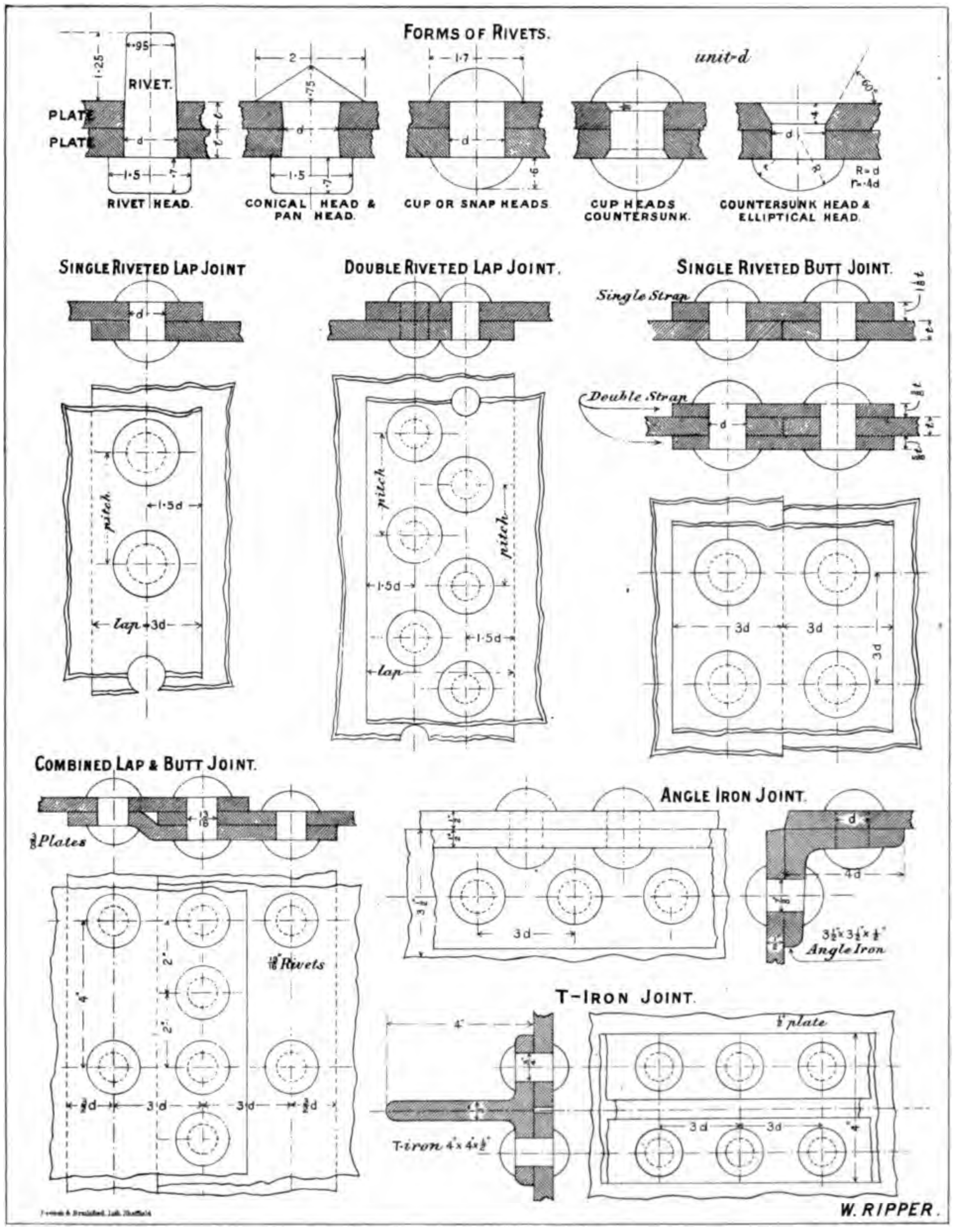


Fig. 1.—Ordinary bolt and nut; draw full size, take d = 1½ ins. Fig. 2.—Connecting rod bolt and nut, showing arrangement for preventing the nut from slacking back; draw full size, take d = 4½ ins.

A = Whitworth thread, p = ½ in. B = Sellers thread, p = 9/16 in. C = Square thread, p = ⅞ in. D = Knuckle thread, p = ⅞ in.
E = Buttress thread, p = 7/16 in. Obtain the dimensions by using the given proportions.

RIVETS AND RIVETED JOINTS.

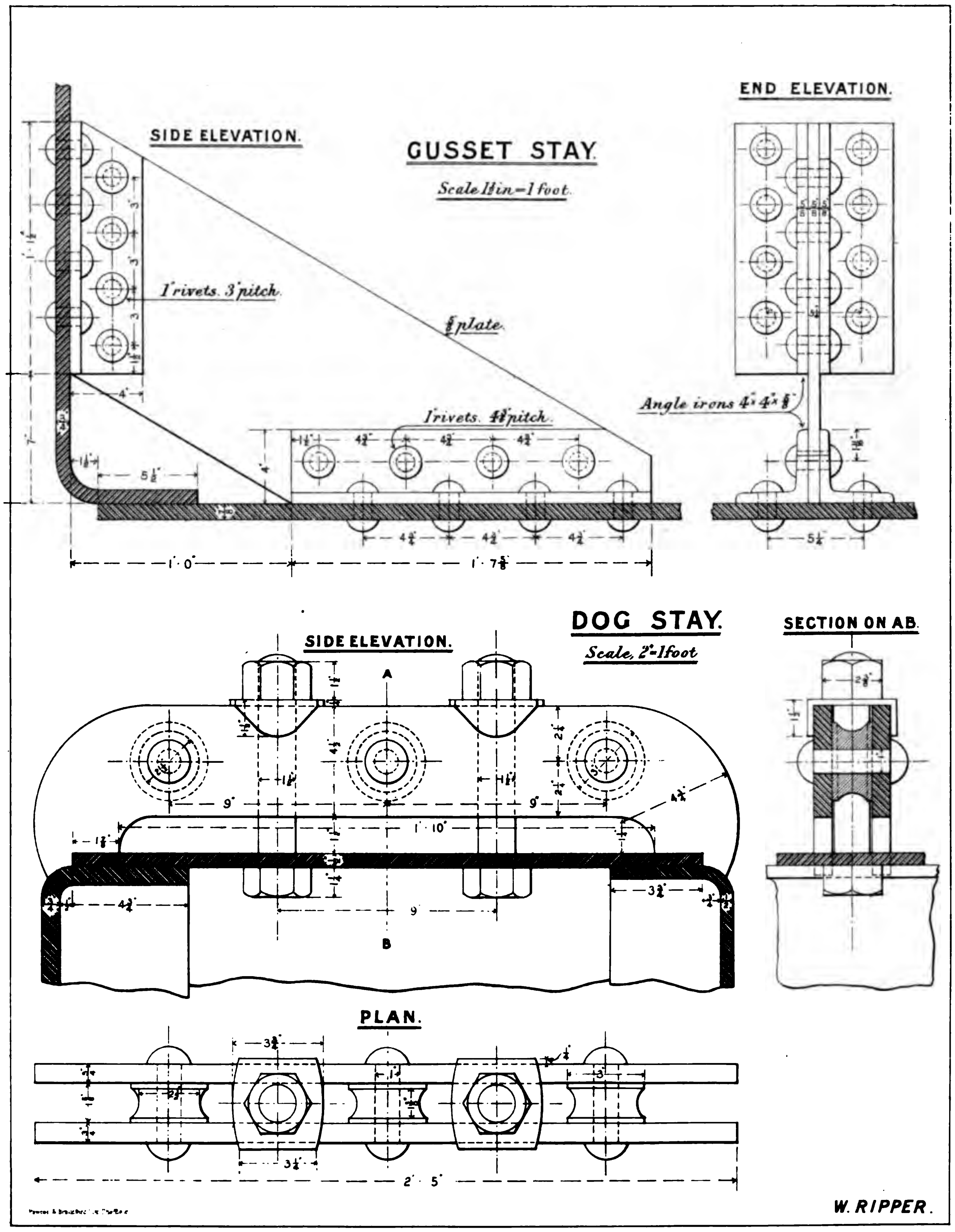
PLATE IX



Draw "Forms of rivets," taking diameter of rivets (d) = 1 in., and thickness of plates (t) = ⅝ in.
Obtain the dimensions by using the given proportions of d.
For dimensions of lap and butt joints see Table, and take thickness of plate = ½ in. Draw full size.

BOILER STAYS.

PLATE X

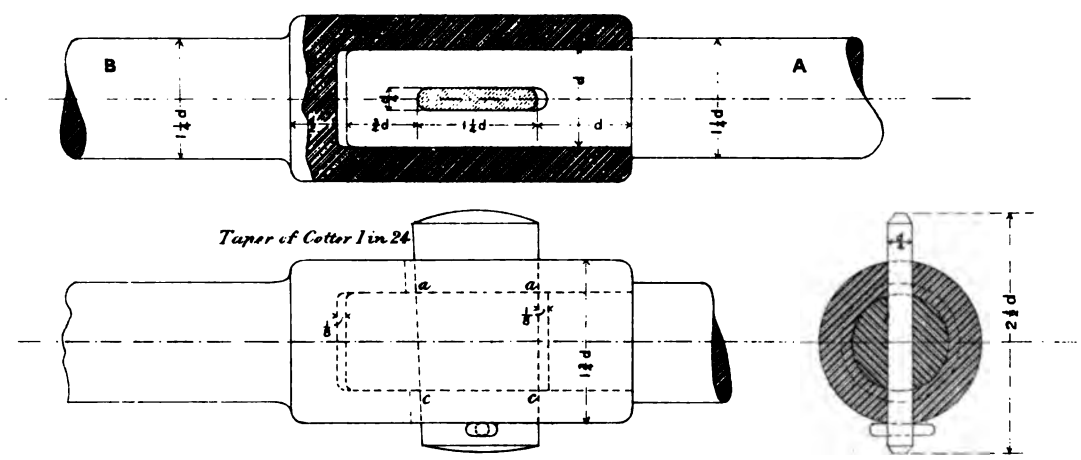


Draw half size. For proportions of hexagon nuts see Plate VIII. and Table IX.
For proportions of rivets see Plate IX.

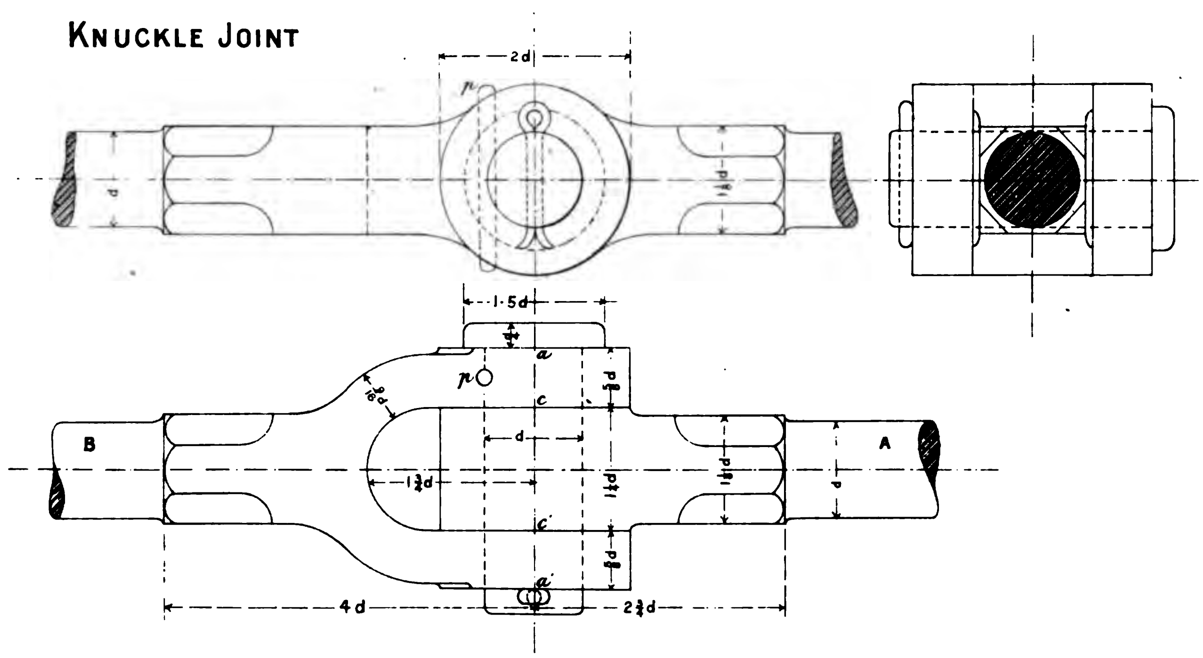
JOINTS AND CONNECTIONS.

PLATE XI

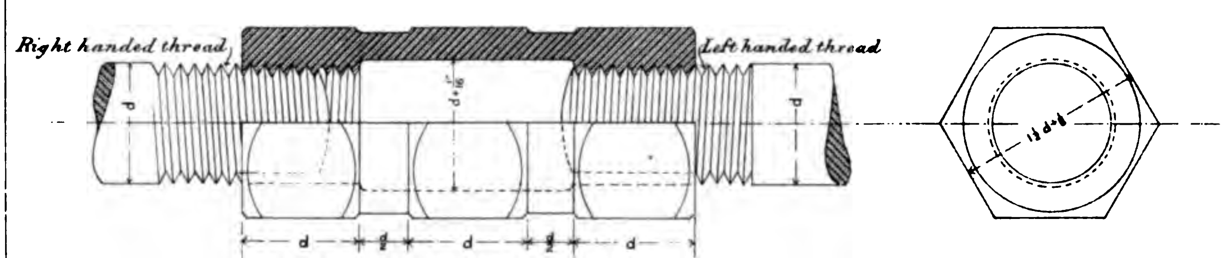
COTTER JOINT



KNUCKLE JOINT



DOUBLE NUT JOINT

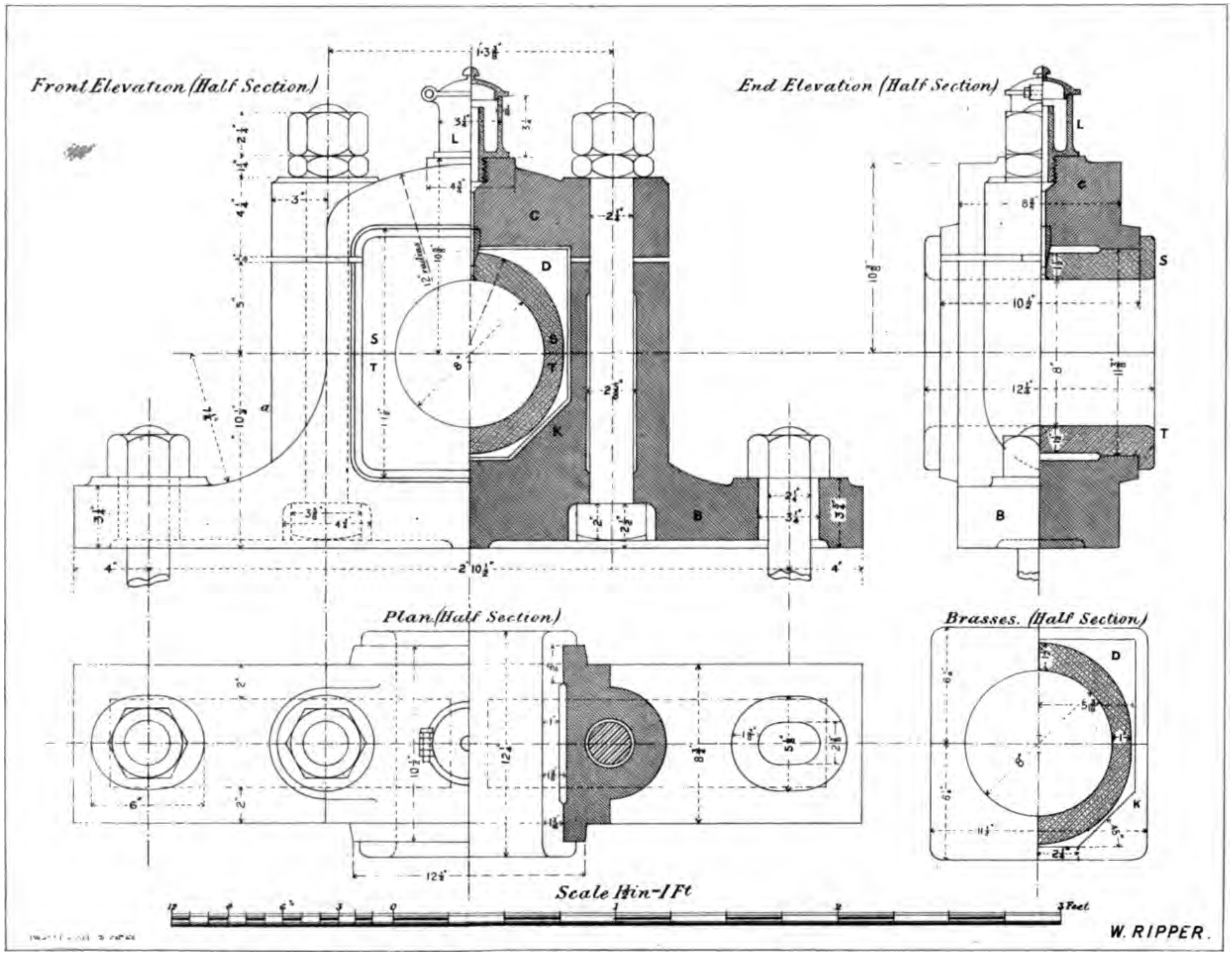


W. RIPPER.

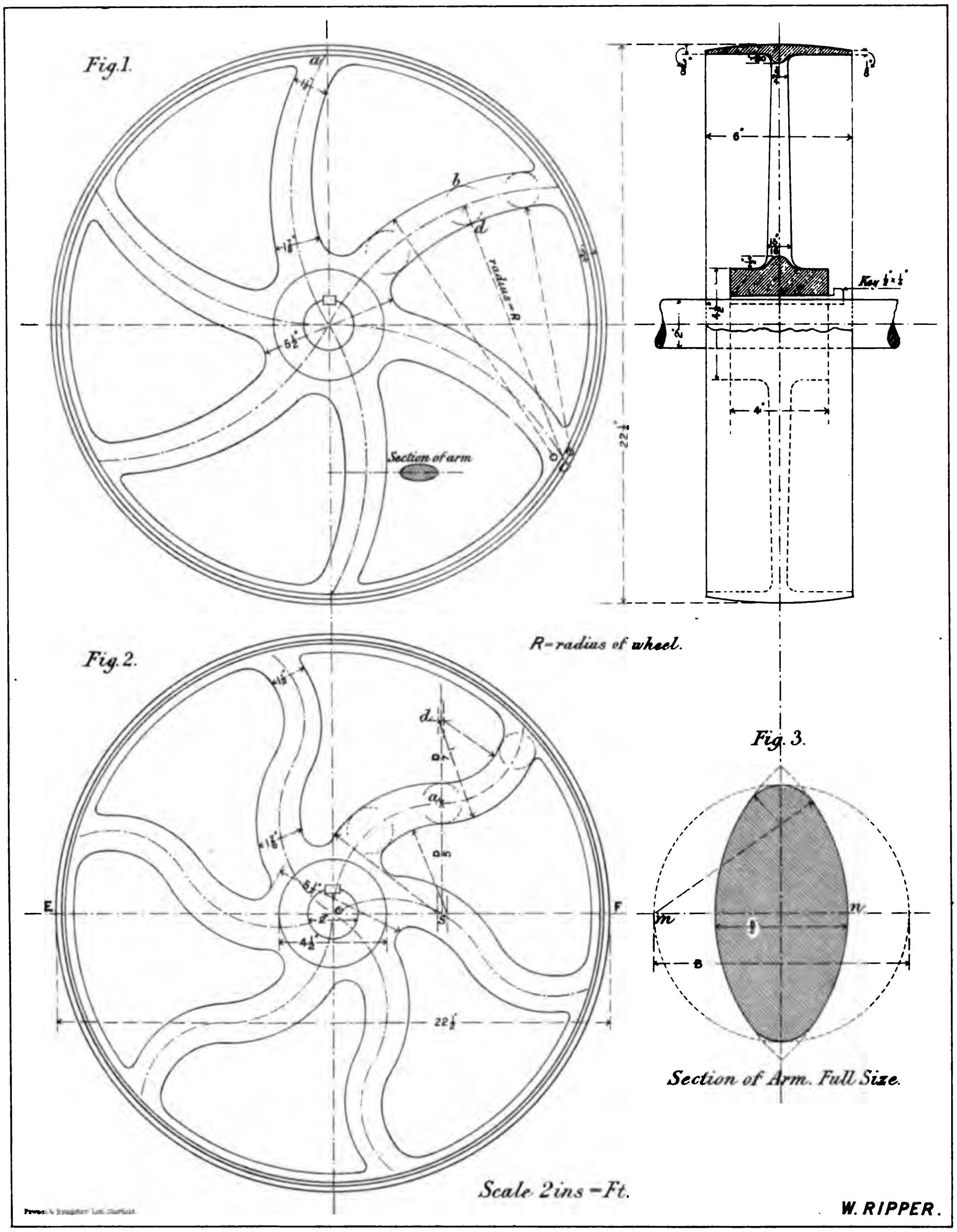
Draw full size each of the above joints when d = 1¼ ins.

PEDESTAL.

PLATE XII



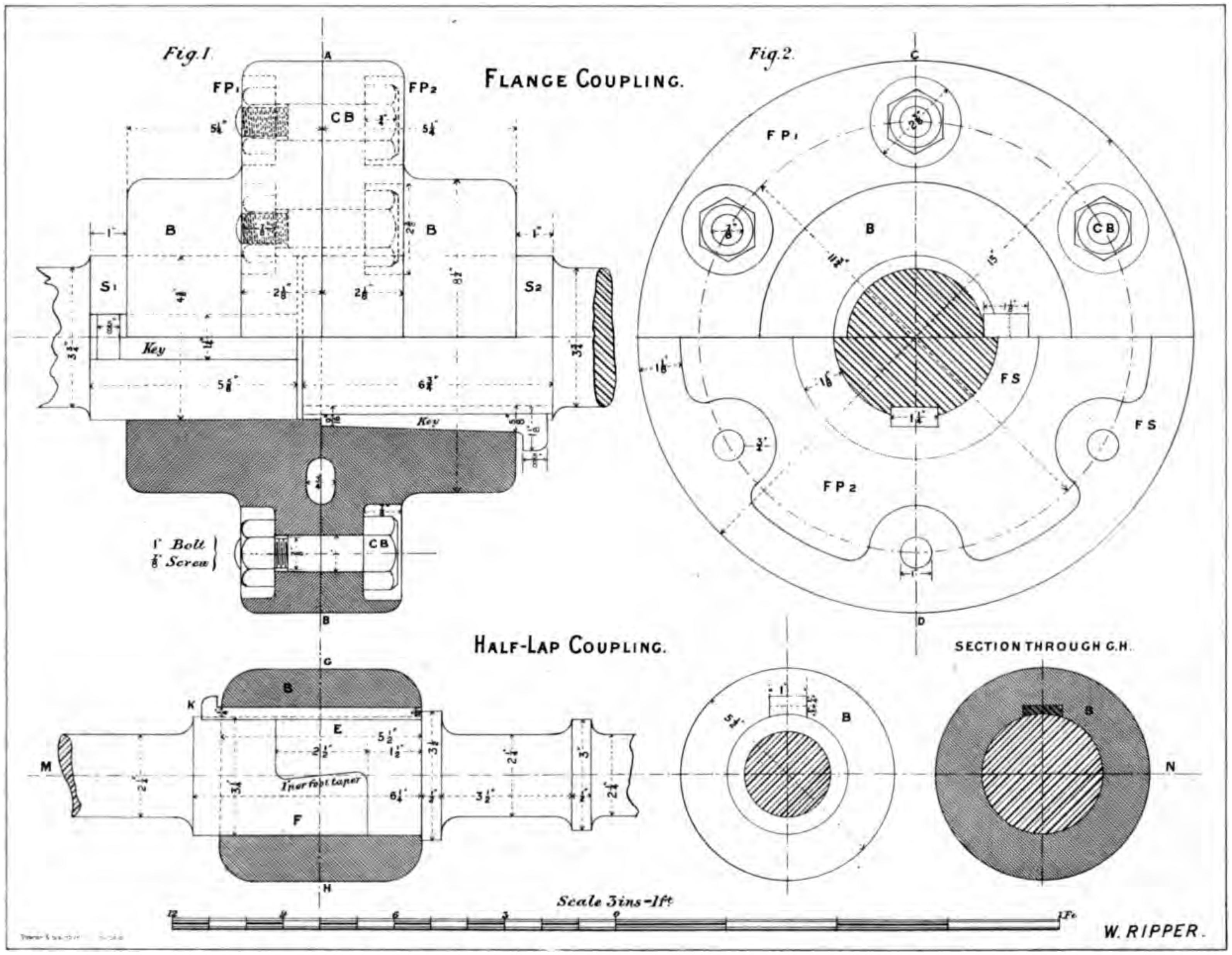
C cap. B base. S and T "brasses" or "steps." D square shoulder. K Octagonal shoulder. L lubricator.
Draw to a scale of 3 ins. = 1 ft. Observe that the views are projected one from the other.



Draw half size.

SHAFT COUPLINGS.

PLATE XIV

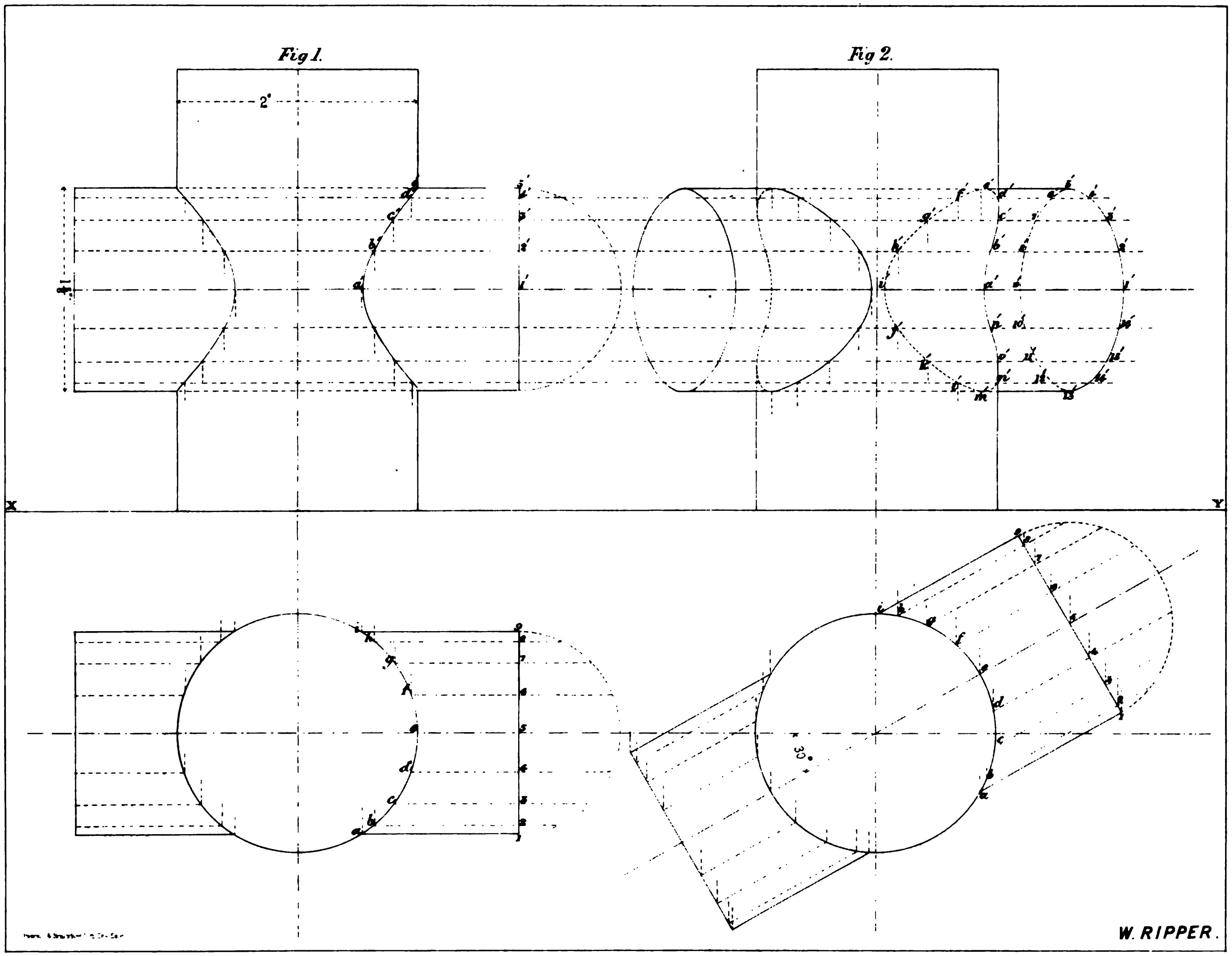


FLANGE COUPLING. *F P1 and F P2 face plates. S1 and S2 shafts. B boss of face plate. C B coupling bolts. F S facing strip.*

HALF-LAP COUPLING. *E and F are ends of shafts over-lapping. B is the cylindrical box secured by a saddle key K.*

Draw half size.

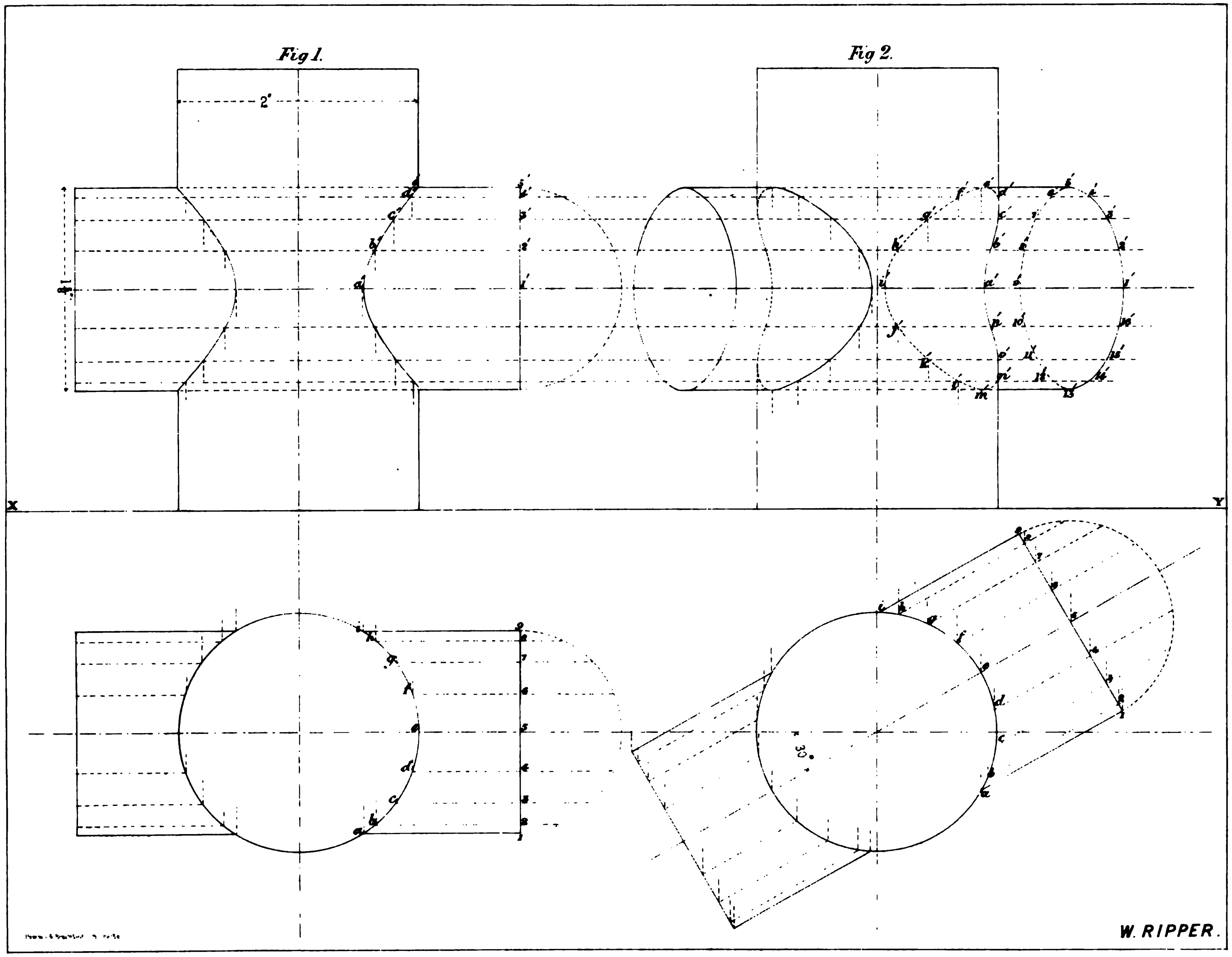
INTERPENETRATIONS OF SOLIDS.



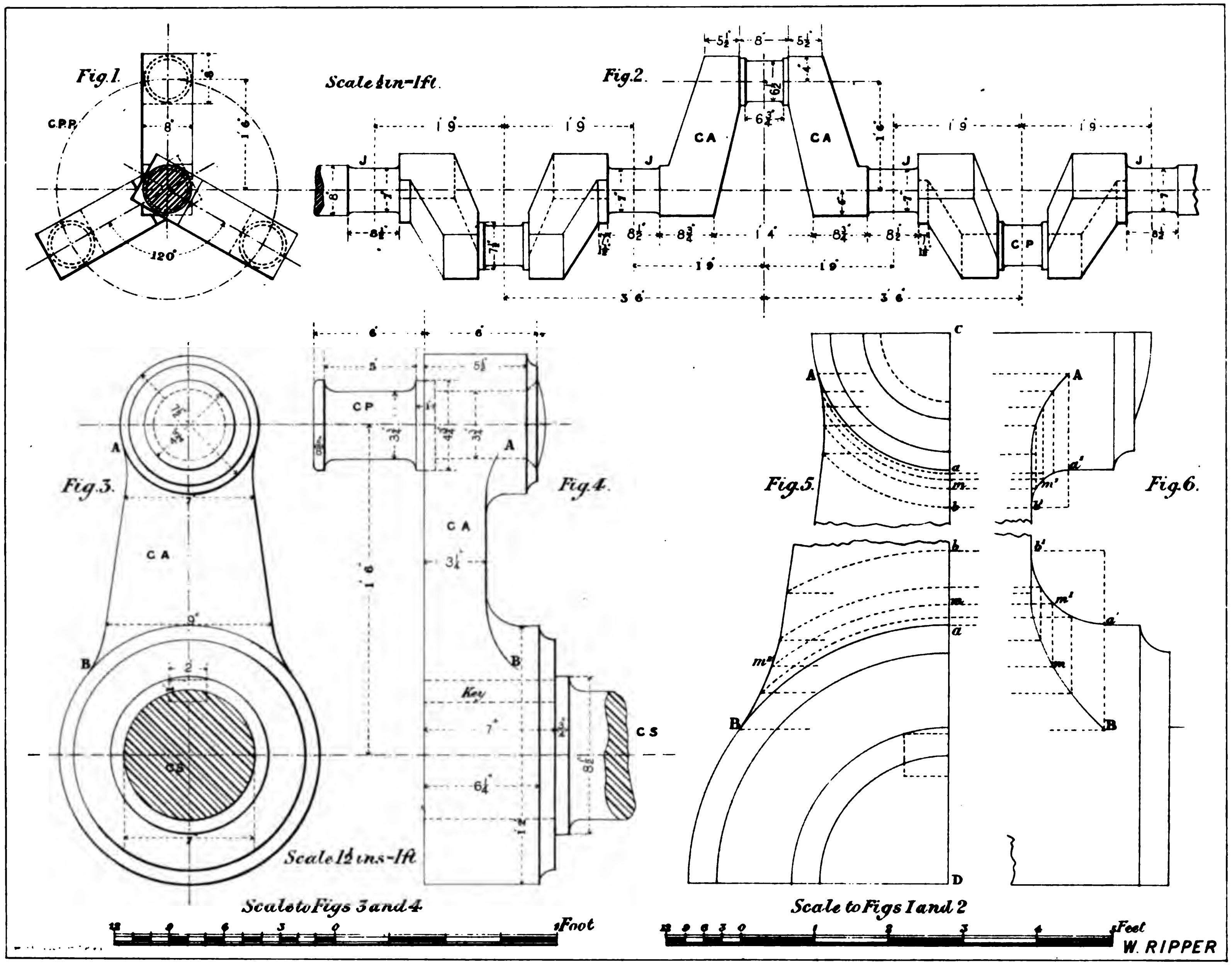
W. RIPPER.

INTERPENETRATIONS OF SOLIDS.

PLATE XV



CRANKS.



C S crank shaft. C P crank pin. J journal. C A crank arm or web. B boss. C P P crank pin path. Draw Figs. 1 and 2 to scale of 1½ ins. to the foot, and Figs. 3 and 4 to scale of 4 ins. to the foot. Figs. 5 and 6 explain the correct method of drawing Fig. 4.

ZEUNER'S VALVE DIAGRAM.

PLATE XXVII

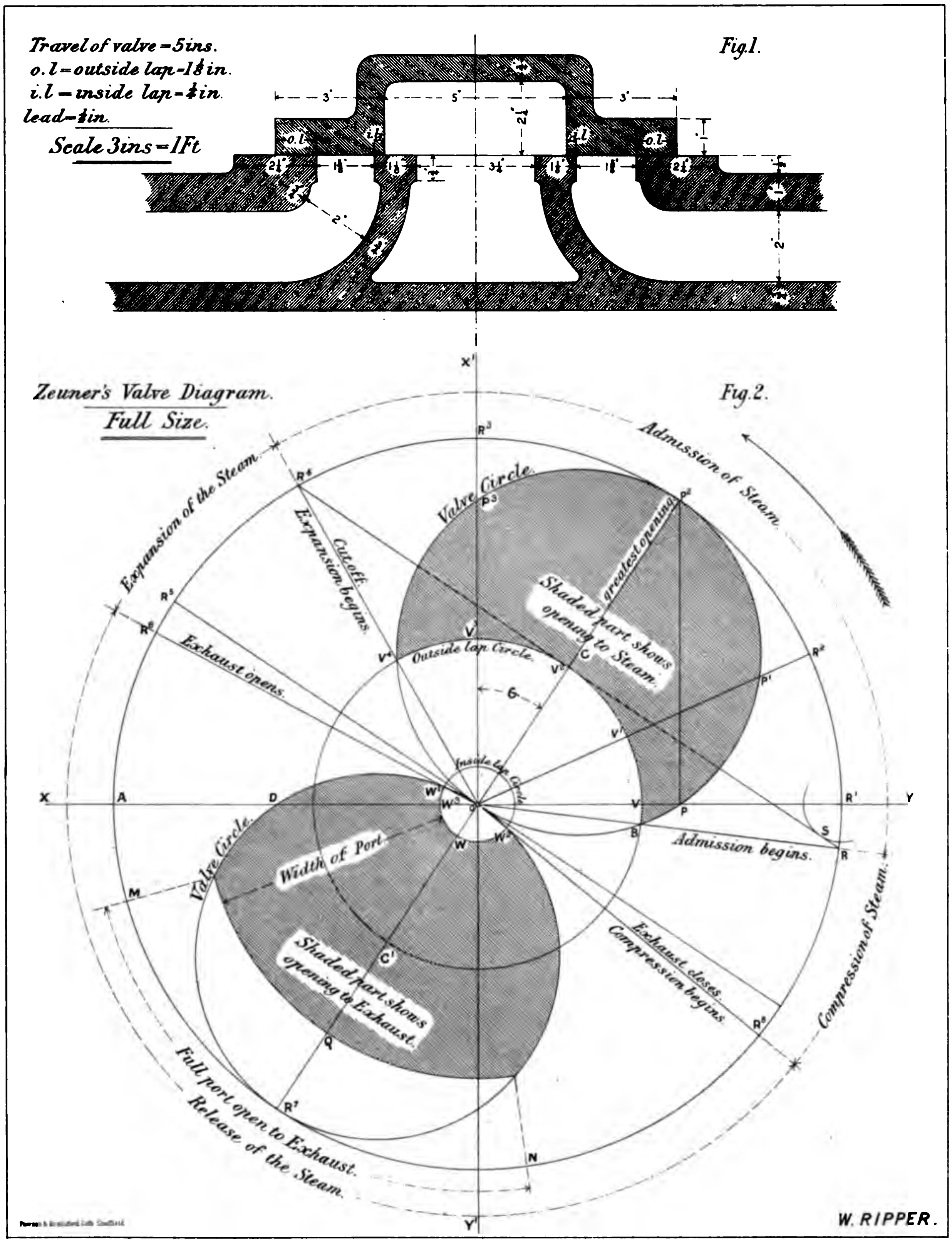


Fig. 1 shows slide valve and ports. Draw full size.
Fig. 2 is the valve diagram for Fig. 1. Draw full size.

LINK-MOTION DIAGRAMS.

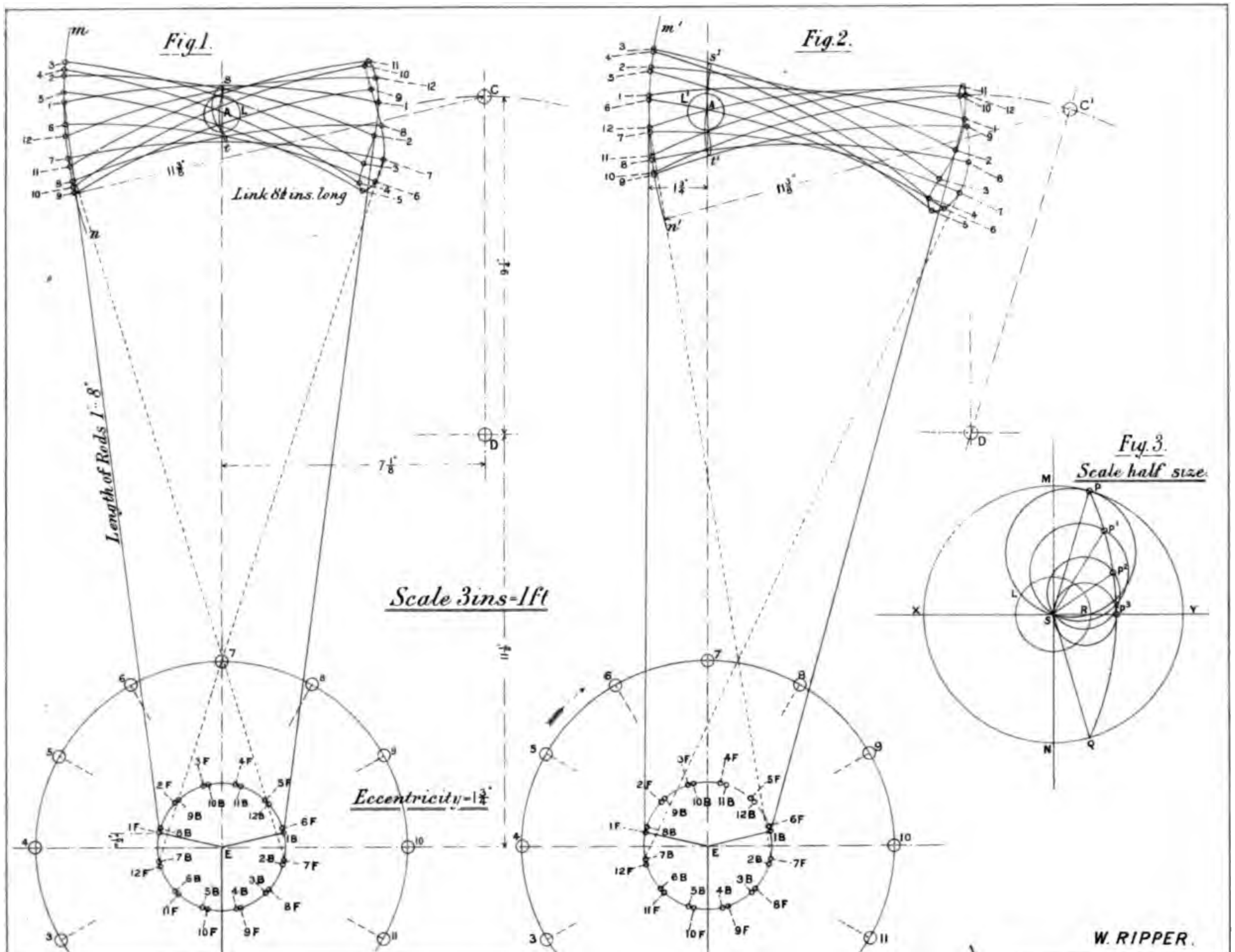
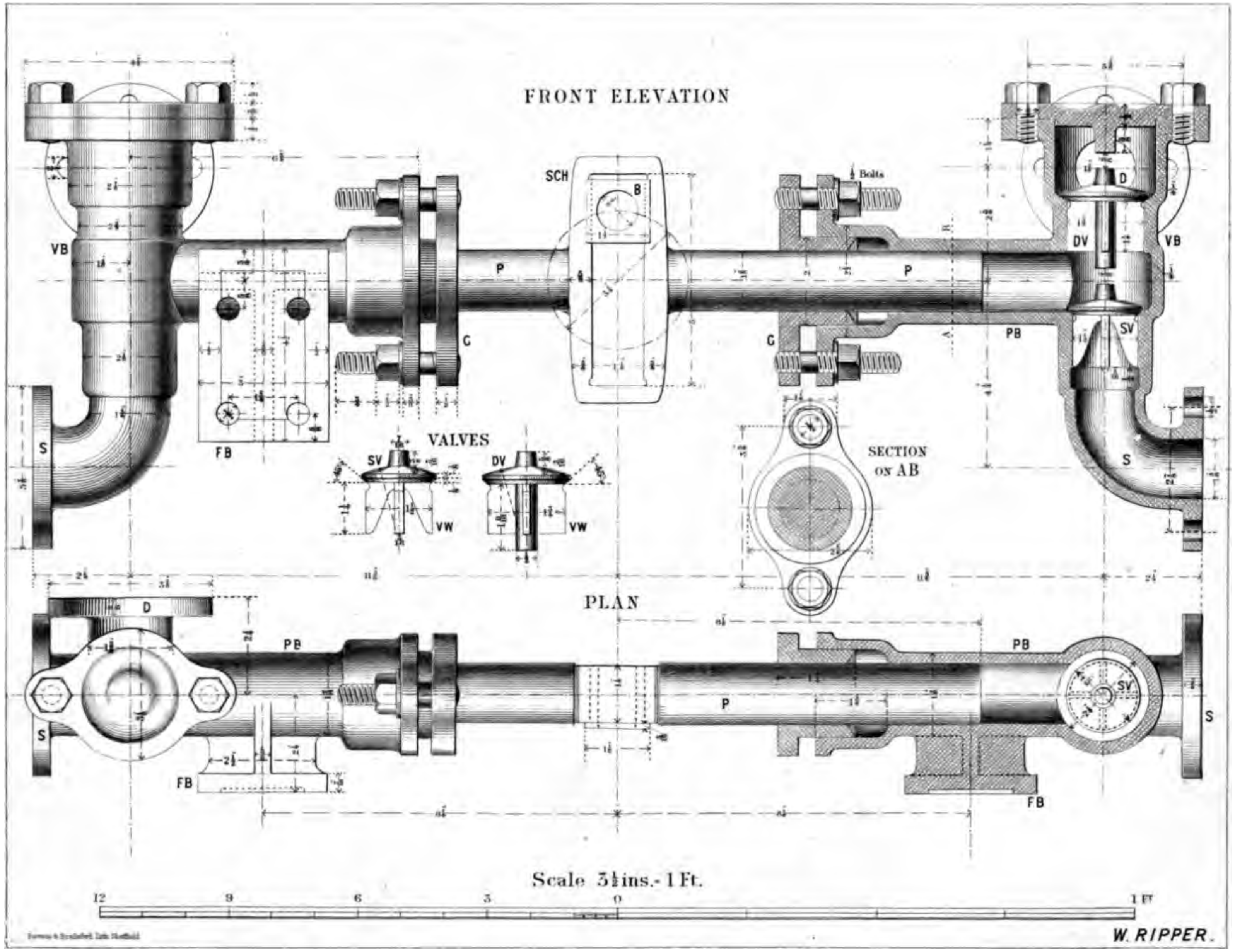


Fig. 1.—Mid gear position. Fig. 2.—Full forward gear position Fig. 3.—Zeuner diagram. Draw full size.

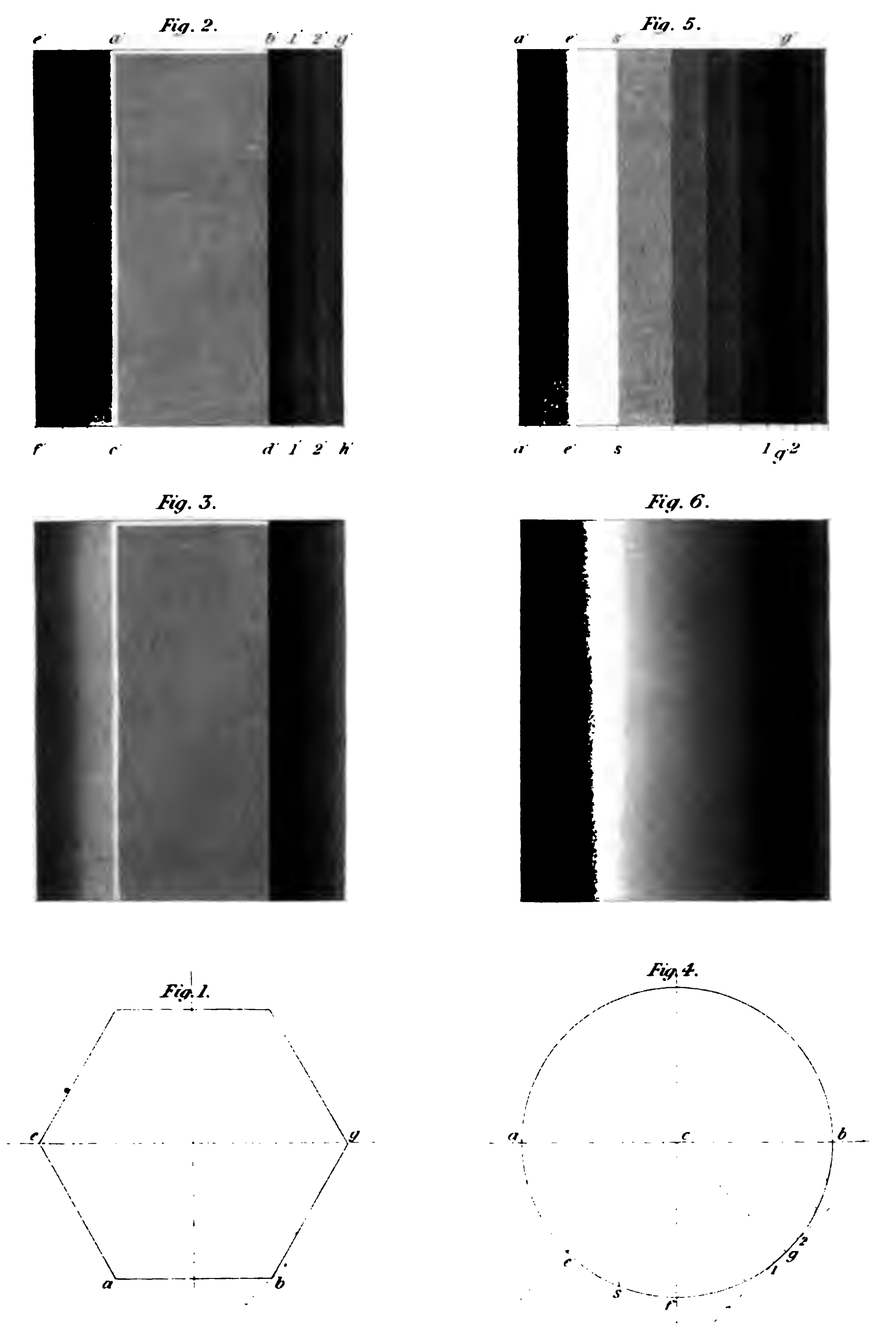
PUMPS. STEAM LAUNCH ENGINES.



V B valve box. S V suction valve. D V delivery valve. P B pump barrel. P plunger. S suction. D delivery. S C H slotted cross head. G gland. B block. F B foot bracket. V W valve webs. Draw to a scale of 9" = 1 ft. Make your own scale.

EXERCISES IN SHADING.

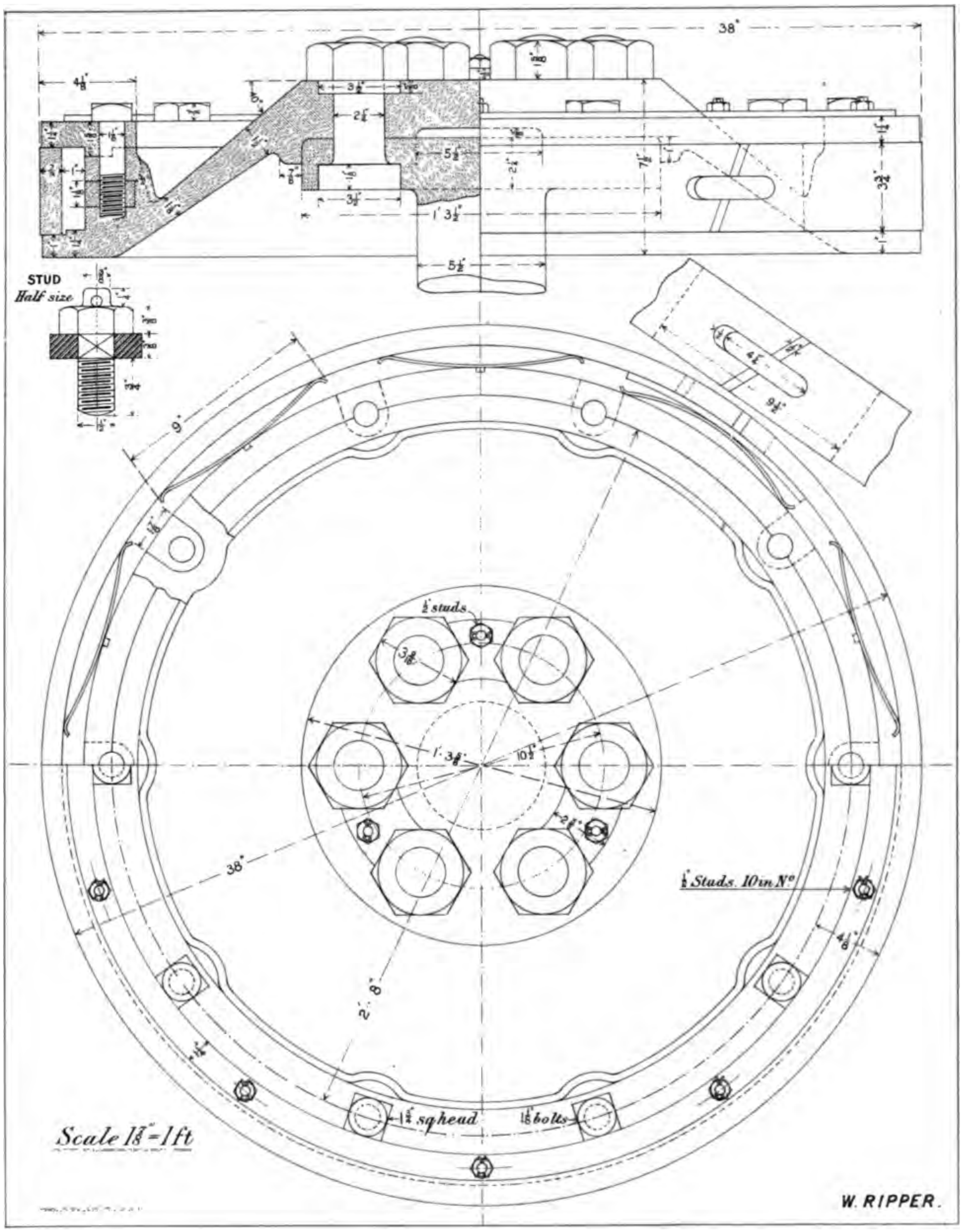
PLATE XXXI



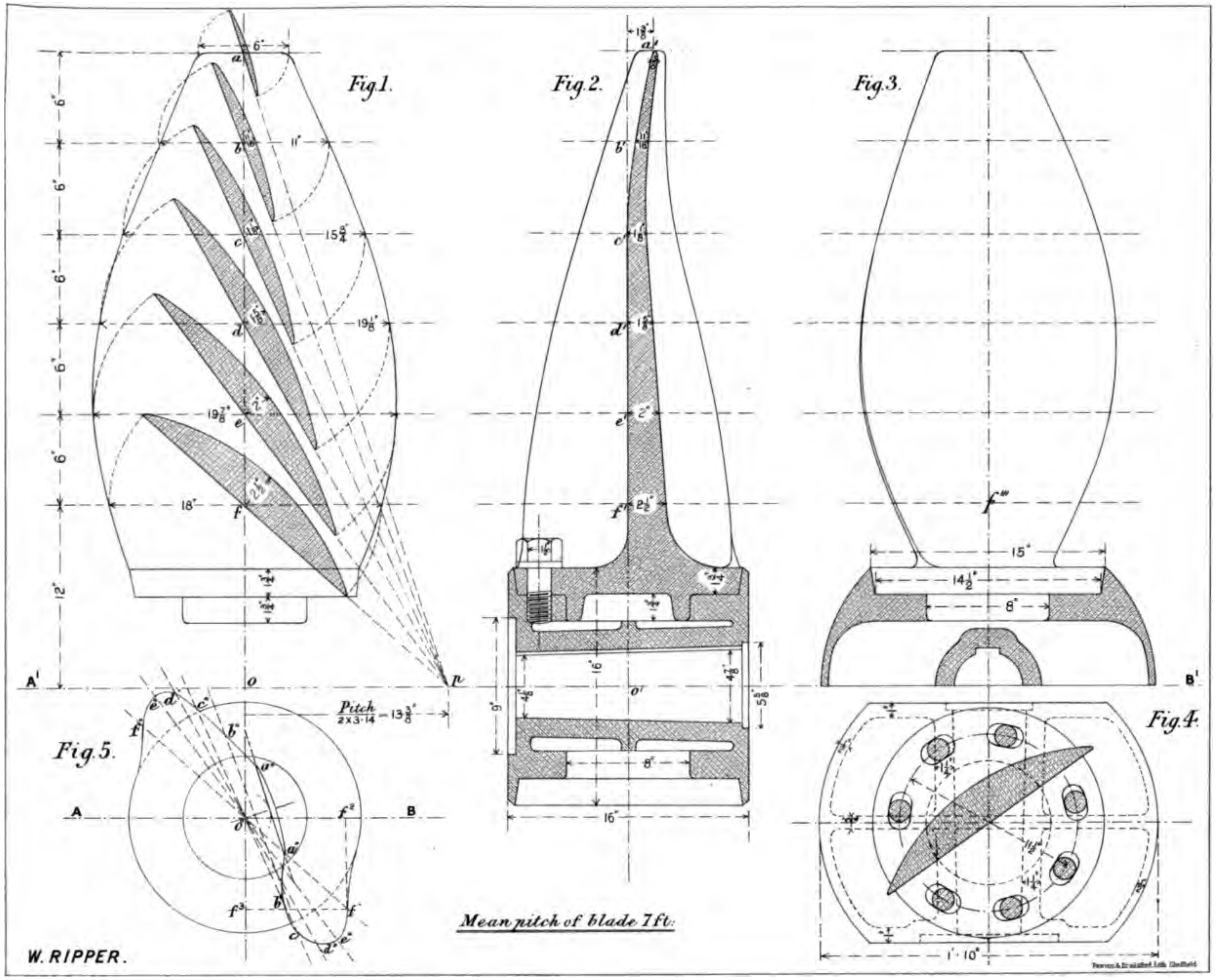
W. RIPPER.

CONICAL STEEL PISTON.

PLATE XXXIIIA

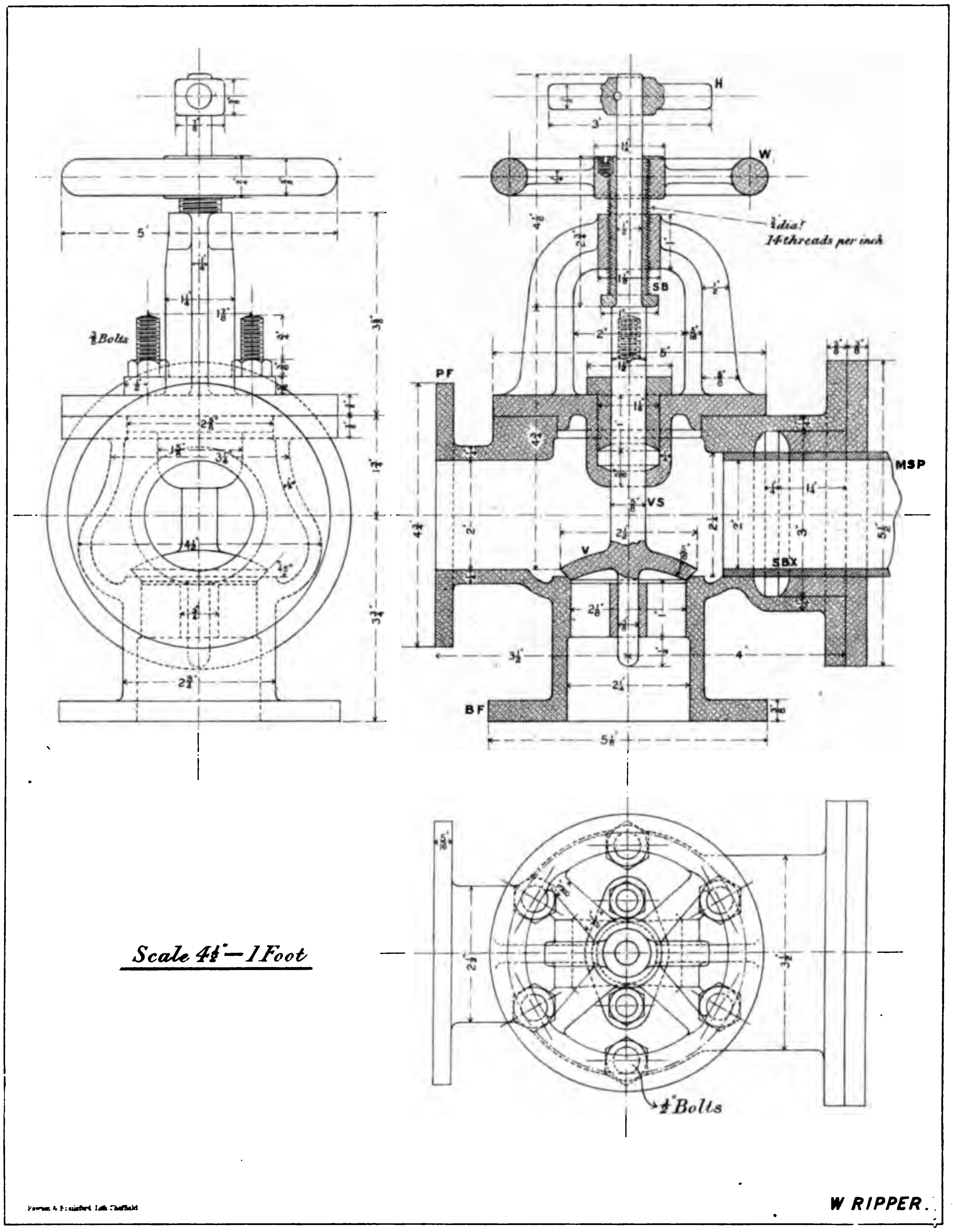


SCREW PROPELLER.



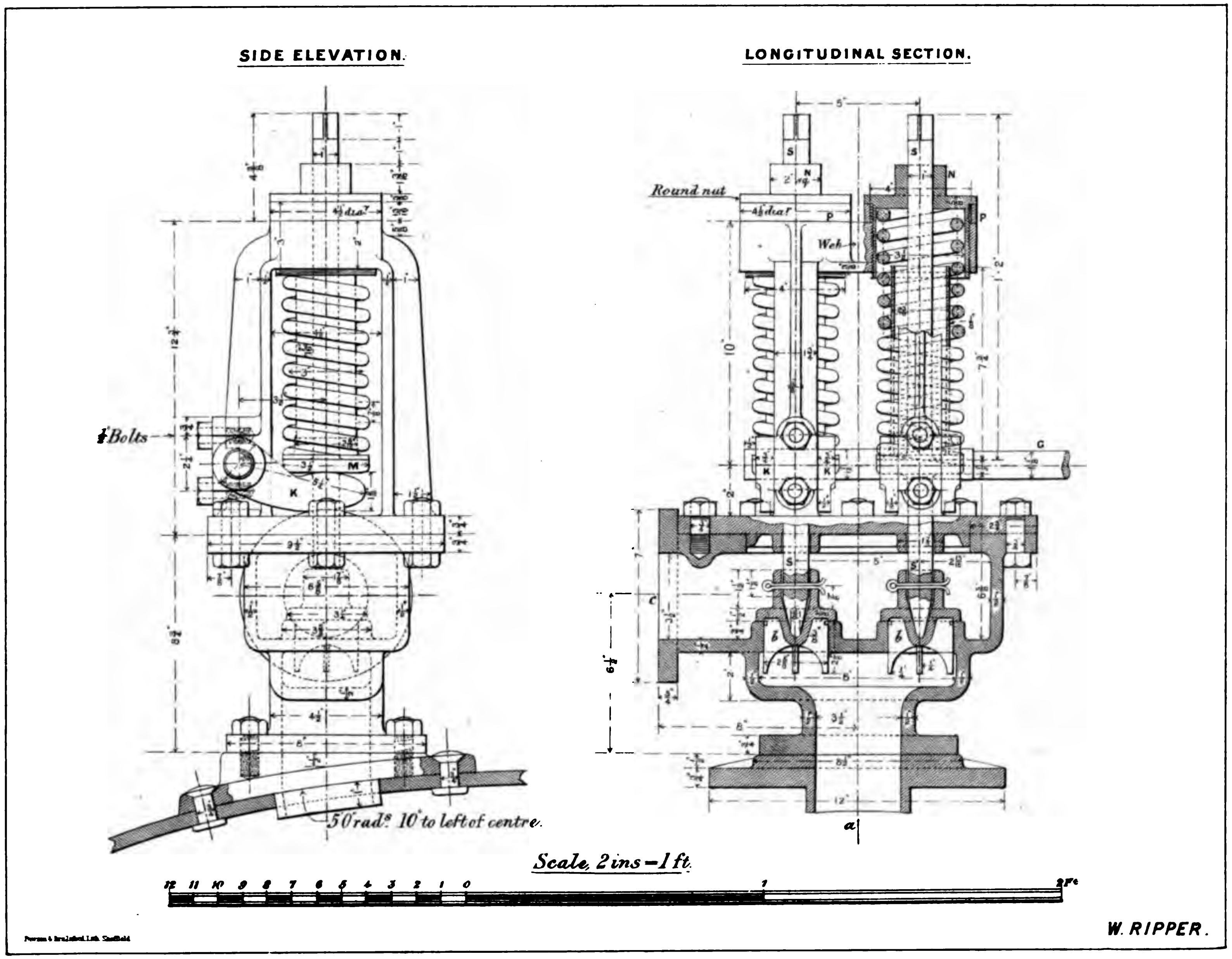
Automatic Stop Valve.

PLATE XXXVI



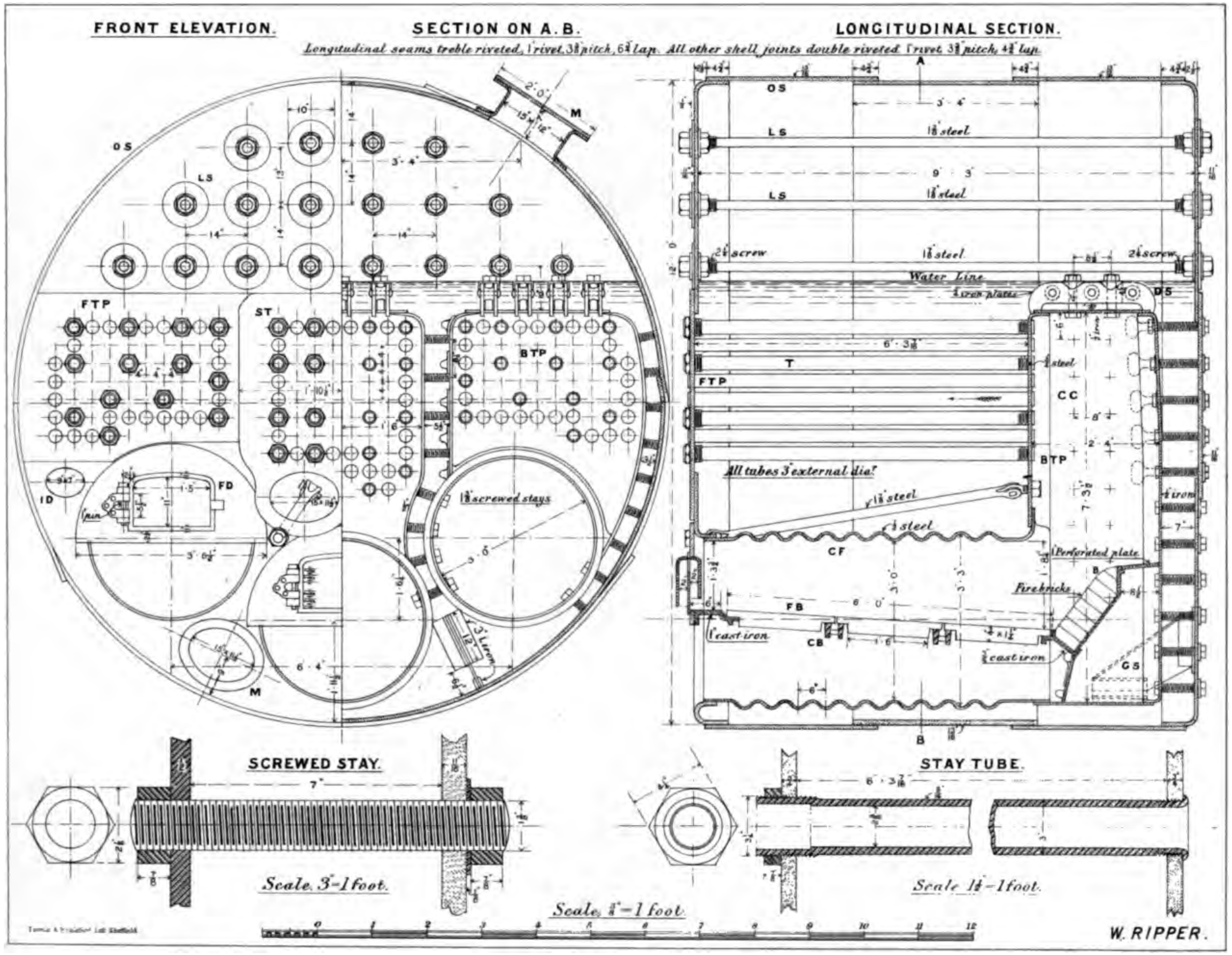
V valve. VS valve spindle. SB screwed bush. W wheel. H handle. SBX stuffing box.
BF flange bolted to front of boiler. PF flange to which copper steam pipe is bolted.
MSP stuffing box end of main steam pipe. Draw full size.

SAFETY VALVES FOR MARINE BOILER.



a communication of valve with boiler. b valves. c outlet. s valve spindle. M collar on valve spindle. N compression nut. P regulating ring or washer. G K arrangement for lifting valve. Draw half size. Work the long way of the paper and draw a plan under longitudinal section.

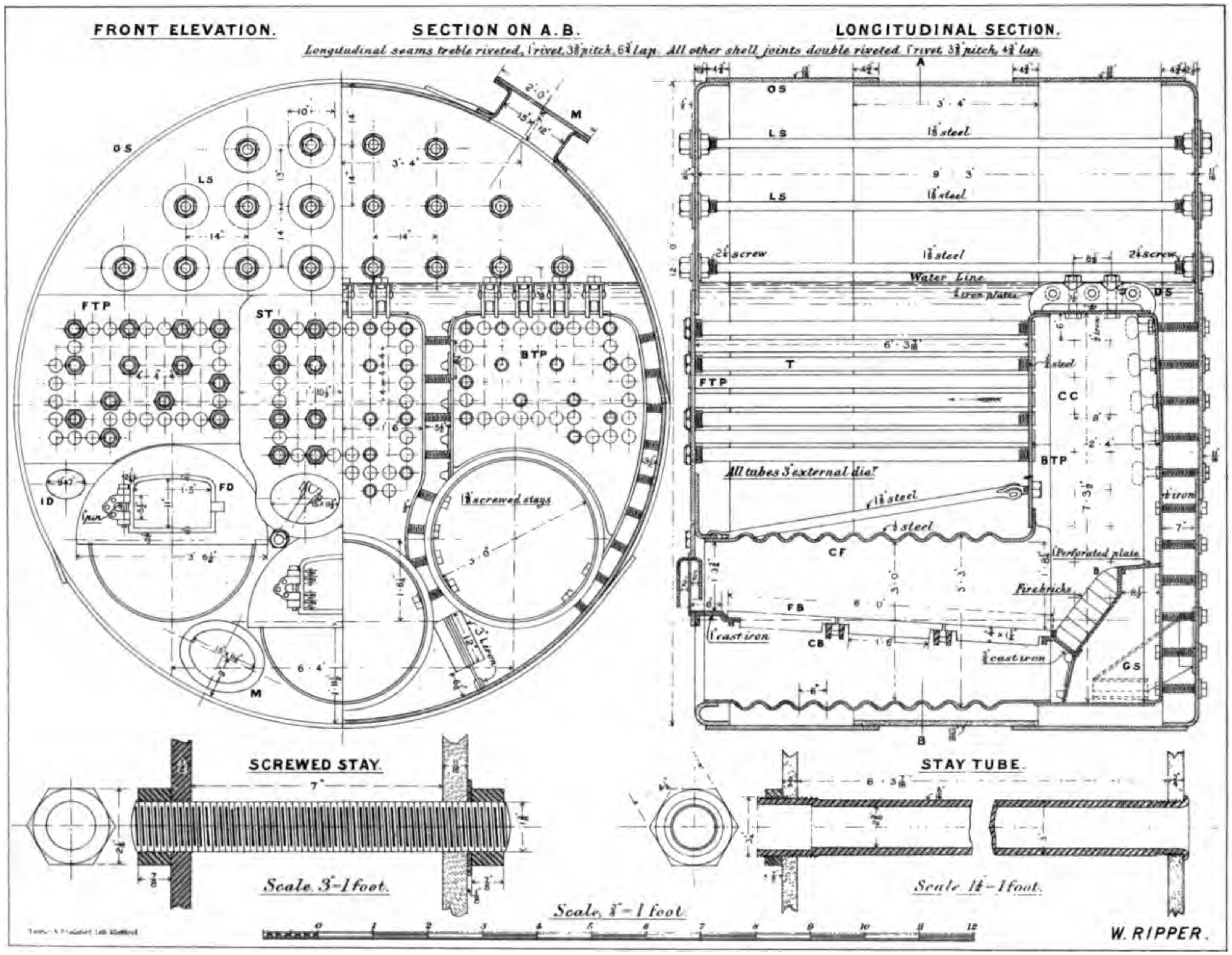
MARINE BOILER.



OS outside shell. CF corrugated furnace flues. CC combustion chamber. FTP front tube plate. BTP back tube plate. T tubes. ST stay tube. DS dog stay (for dimensions see Plate X.) LS longitudinal stay. FD furnace door. FB furnace bars. CB cross bearer. B fire-brick bridge. M manhole. ID inspection door. GS gusset stay.

MARINE BOILER.

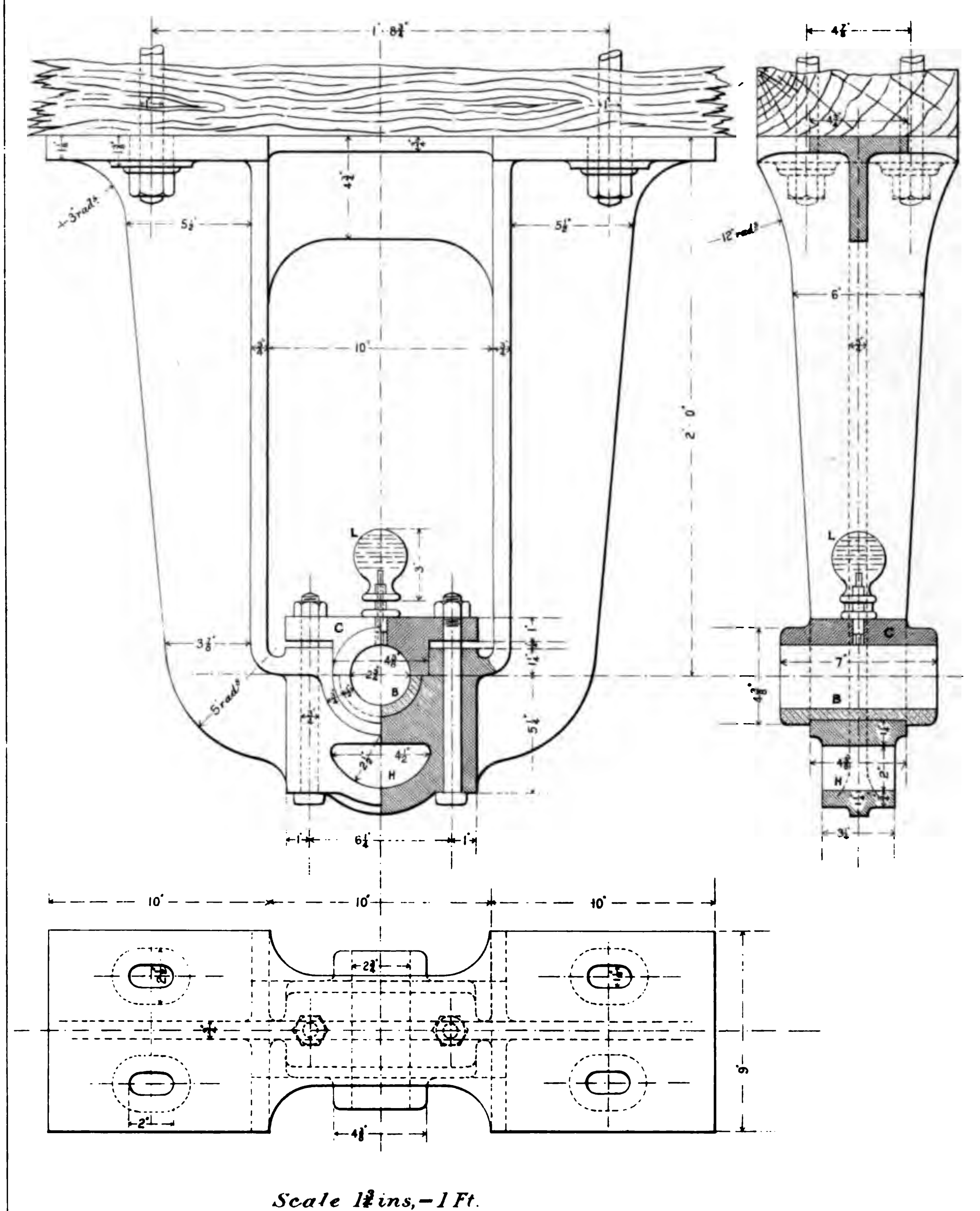
PLATE XXXVIII



O S outside shell. C F corrugated furnace flues. C C combustion chamber. F T P front tube plate. B T P back tube plate. T tubes. S T stay tube. D S dog stay (for dimensions see Plate X.) L S longitudinal stay. F D furnace door. F B furnace bars. C B cross bearer. B fire-brick bridge. M manhole. I D inspection door. G S gusset stay.

HANGER FOR SHAFTING.

PLATE XXXIX



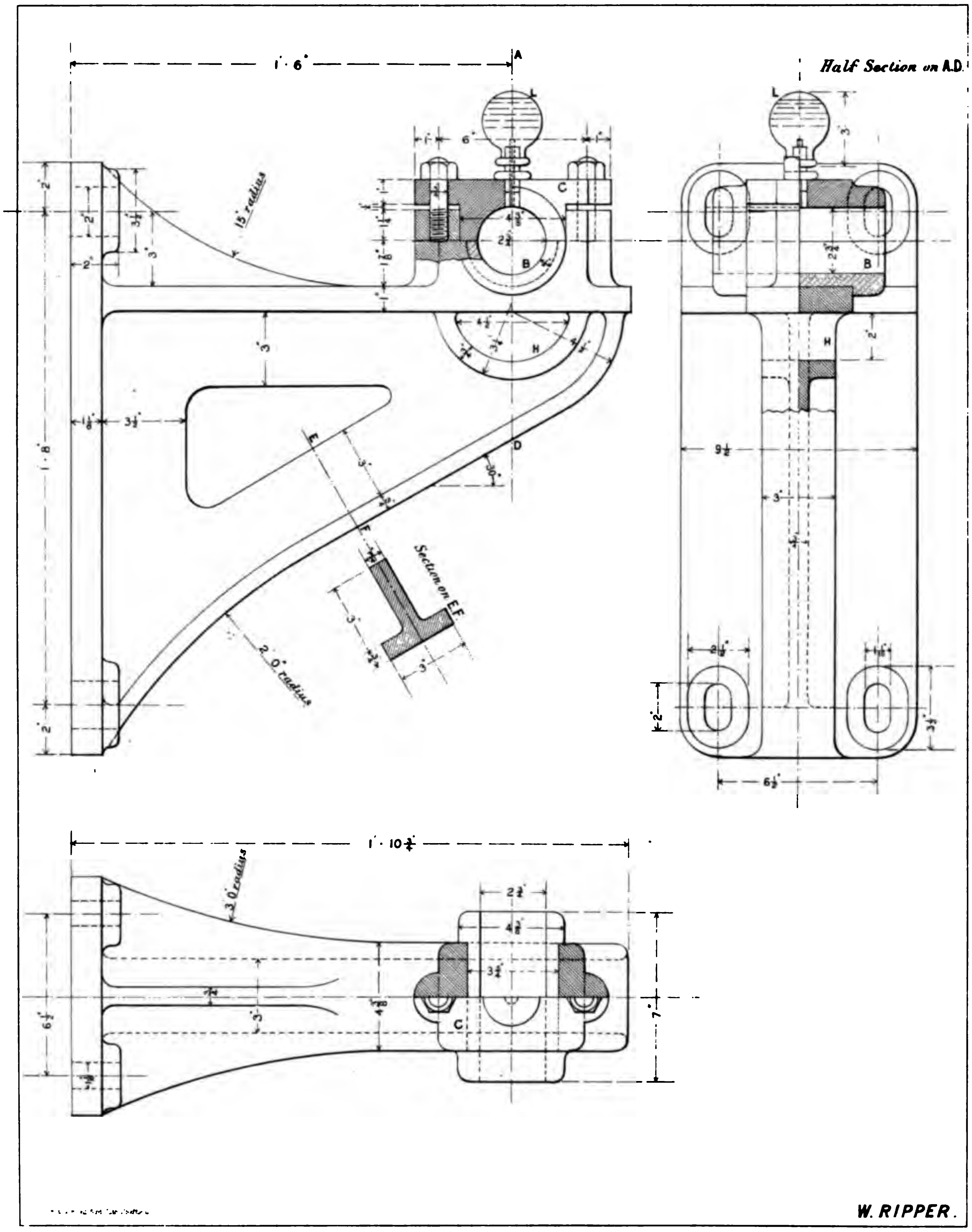
Scale 1½ ins, – 1 Ft.

W. RIPPER.

B bearing for shaft. C cap. L lubricator. H hole for waste oil can. Draw 6 ins. = 1 ft.

WALL BRACKET FOR SHAFTING.

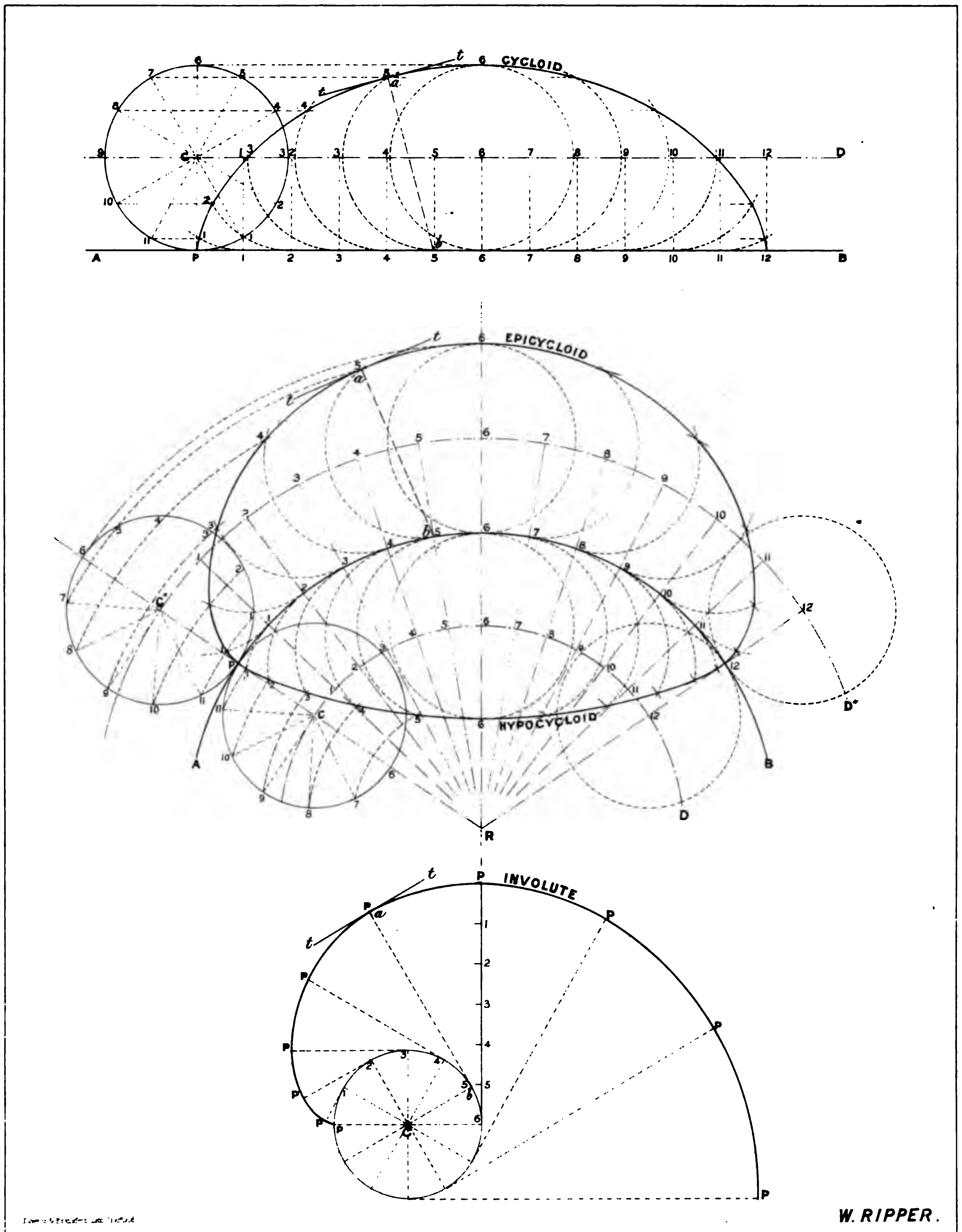
PLATE XL



B bearing for shaft. C cap. L lubricator. H hole for waste oil can. Draw 6 ins. = 1 ft.

CURVES.

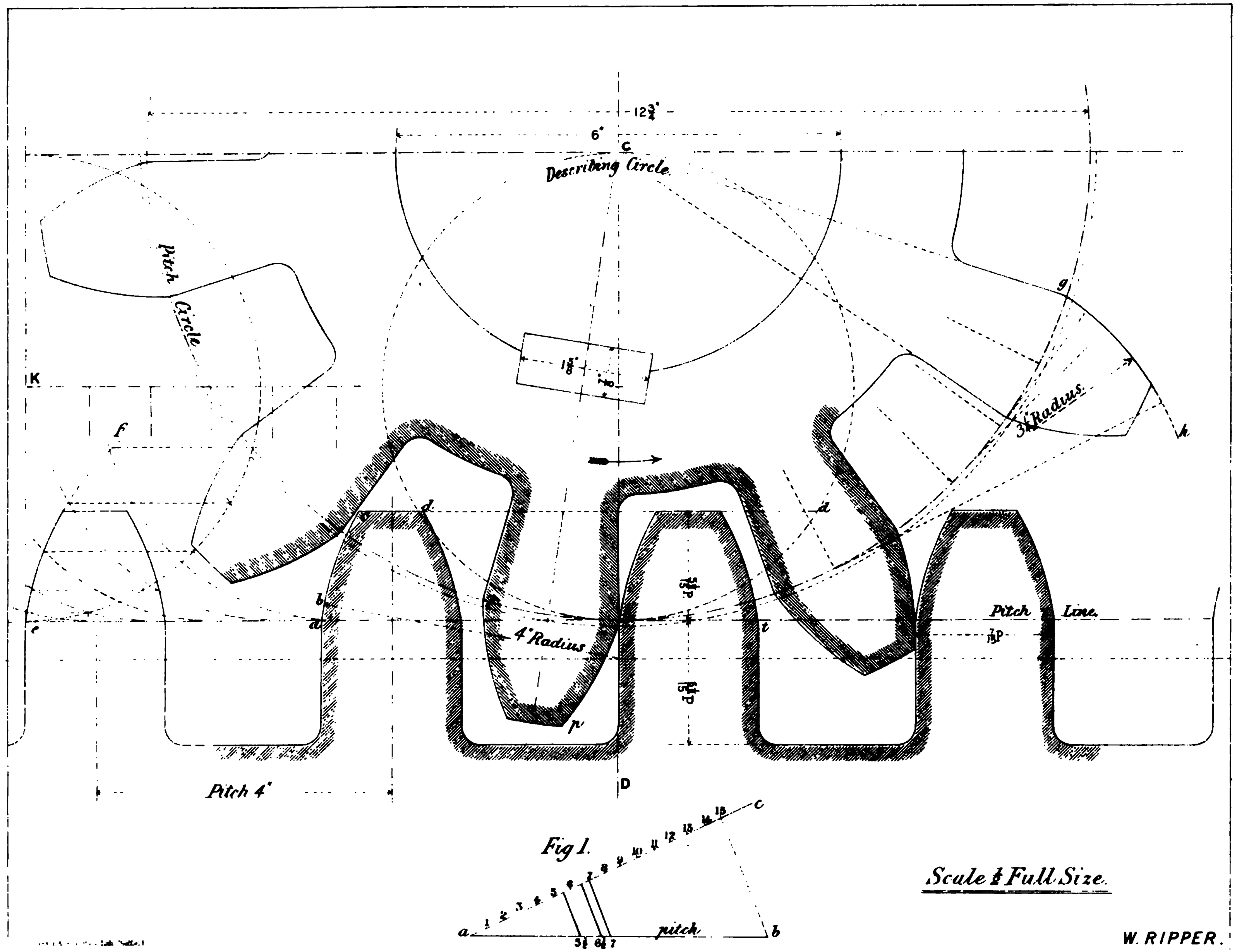
PLATE XLI



W. RIPPER.

RACK AND PINION IN GEAR.

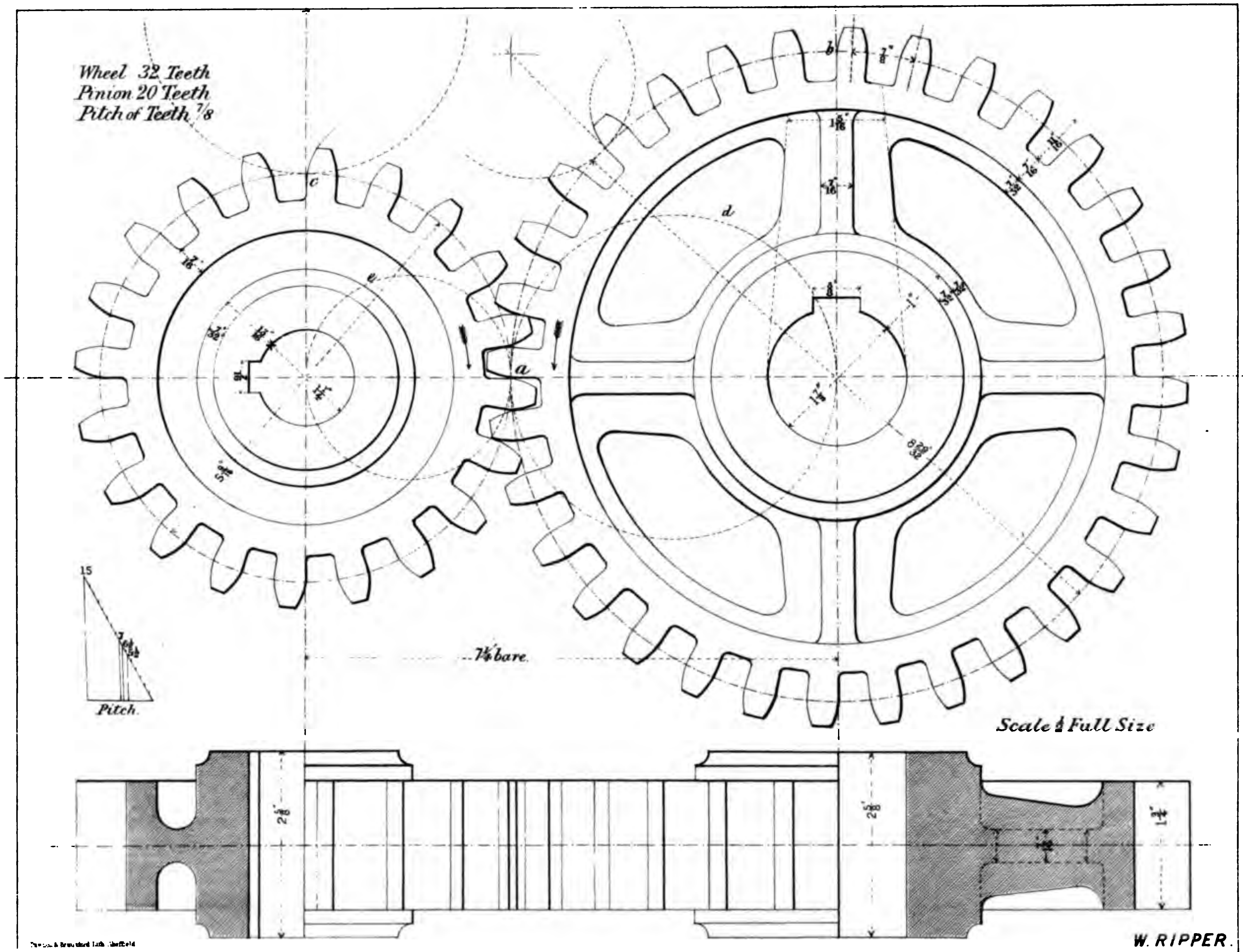
PLATE XLII



Draw full size.

Wheel and Pinion in Gear.

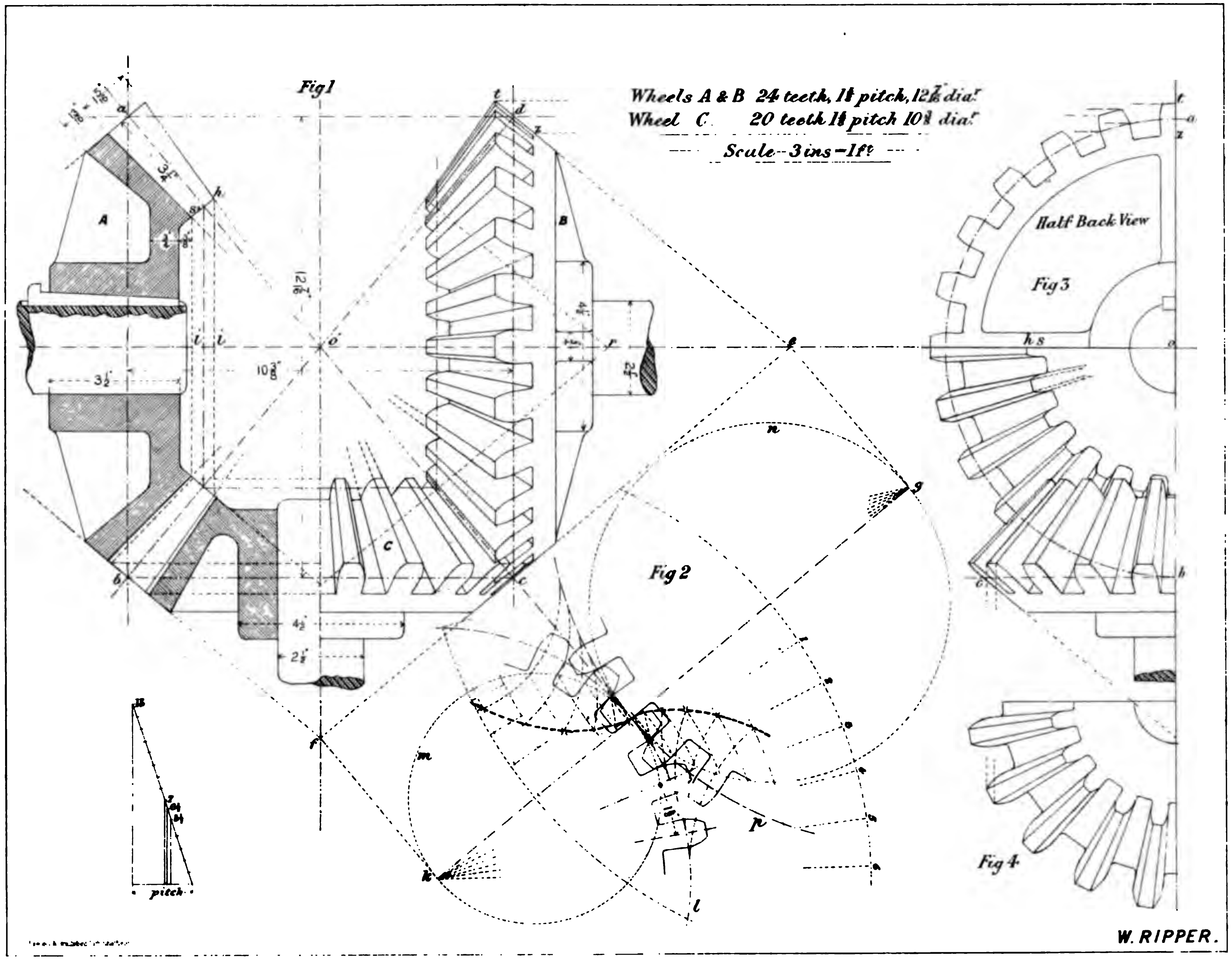
PLATE XLIII



Draw full size.

Bevel Wheels.

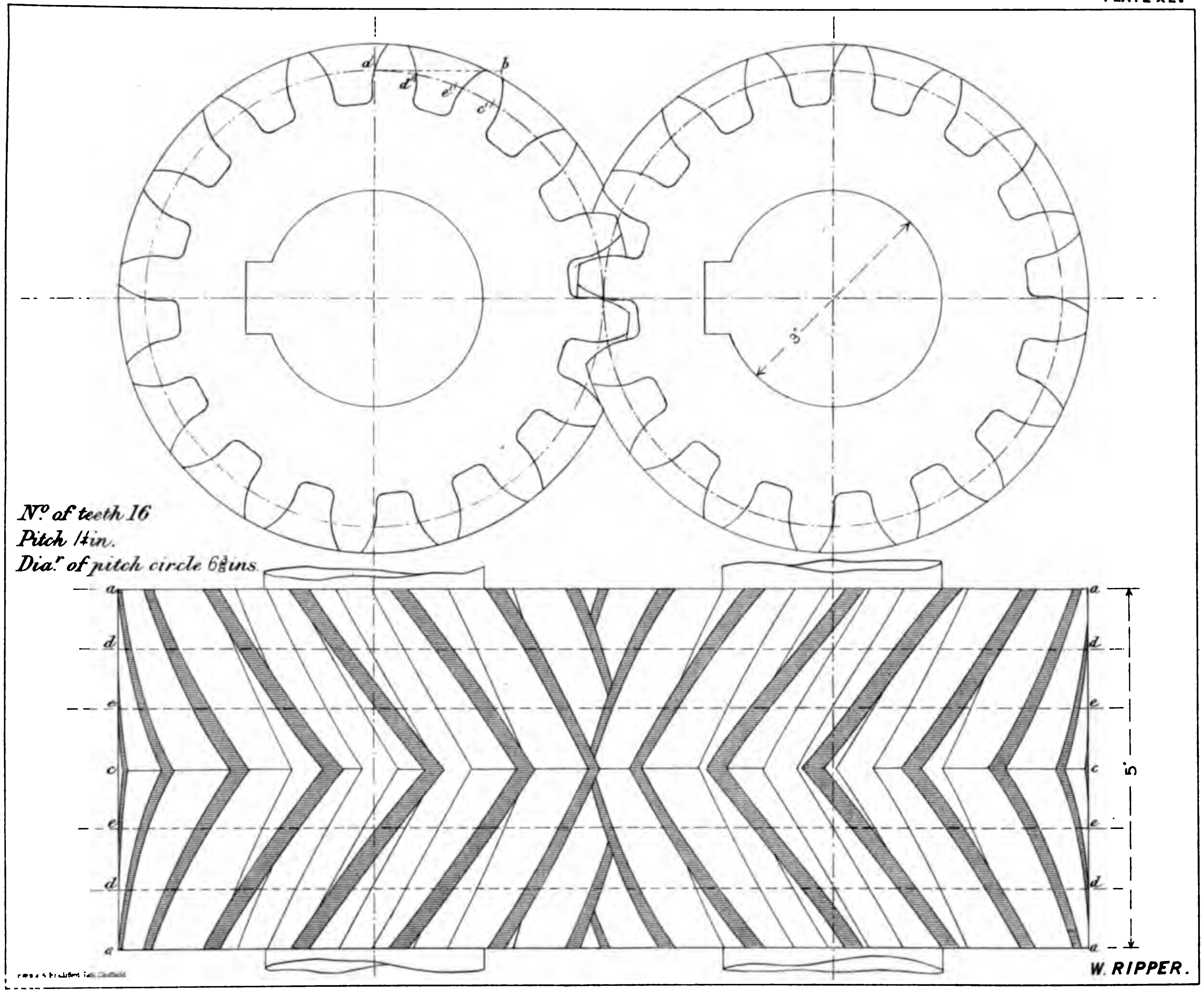
PLATE XLIV



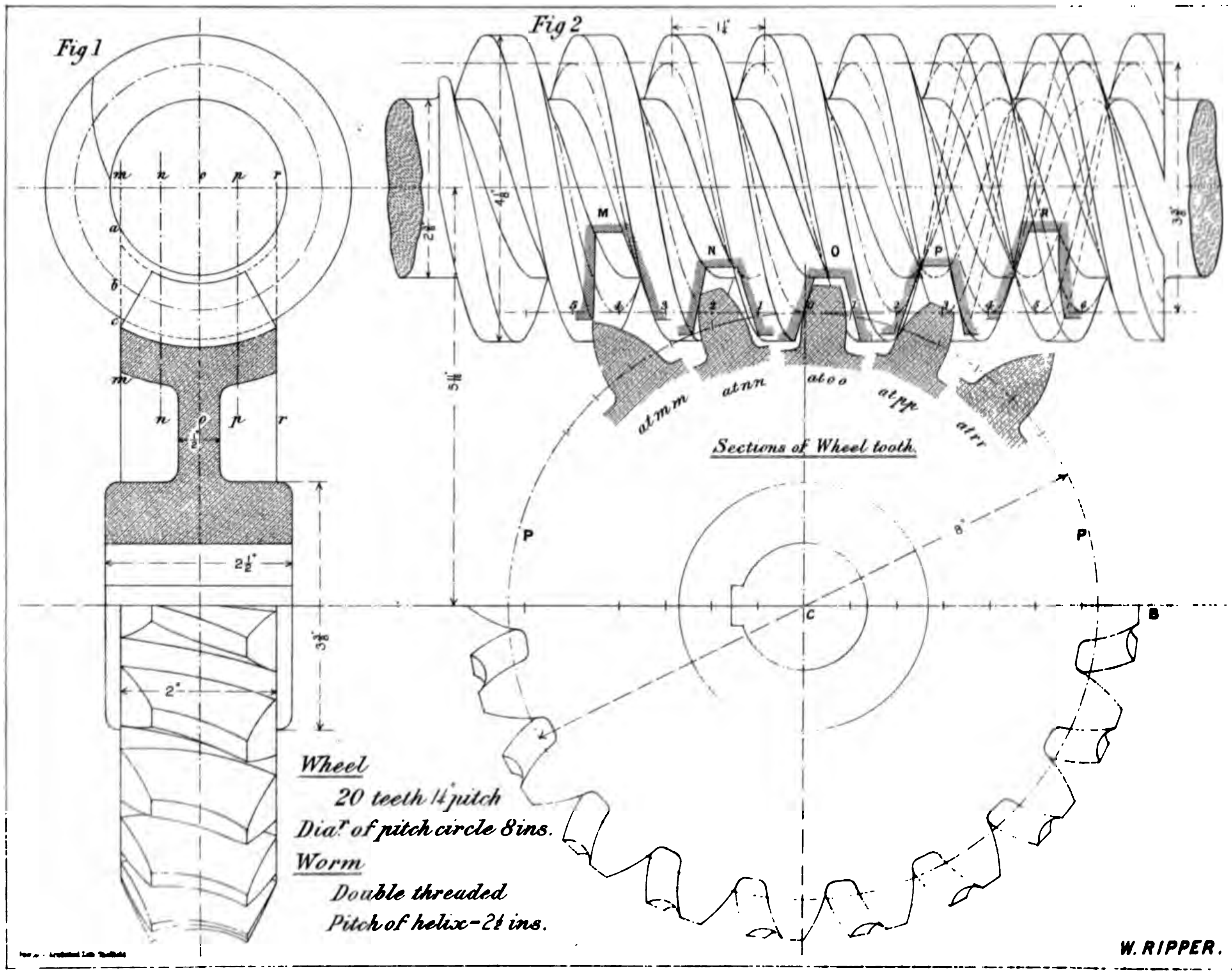
Draw to a scale of 9 ins. = 1 foot.

HELICAL GEARING.

PLATE XLV

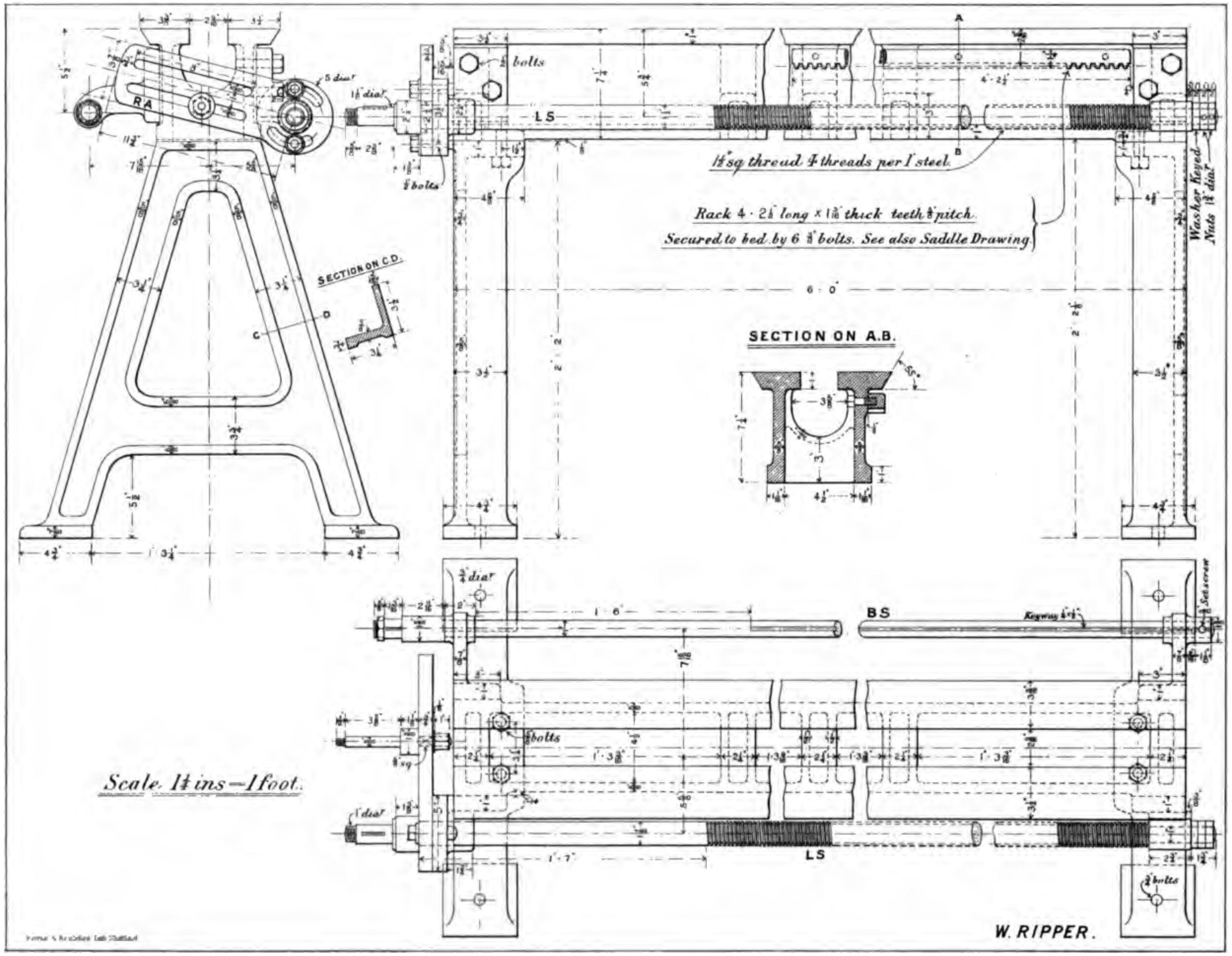


WORM AND WHEEL.



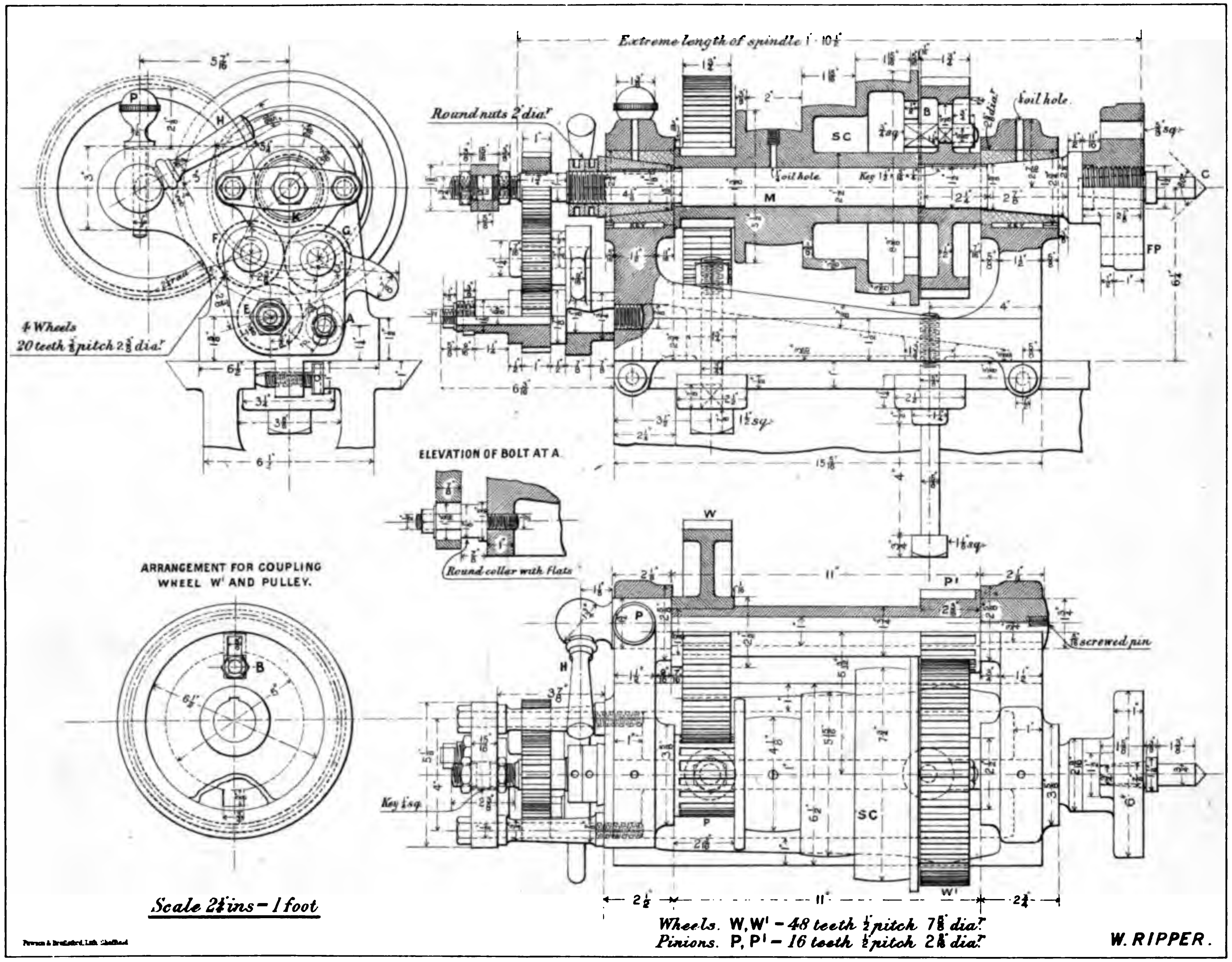
LATHE BED.

PLATE XLVII



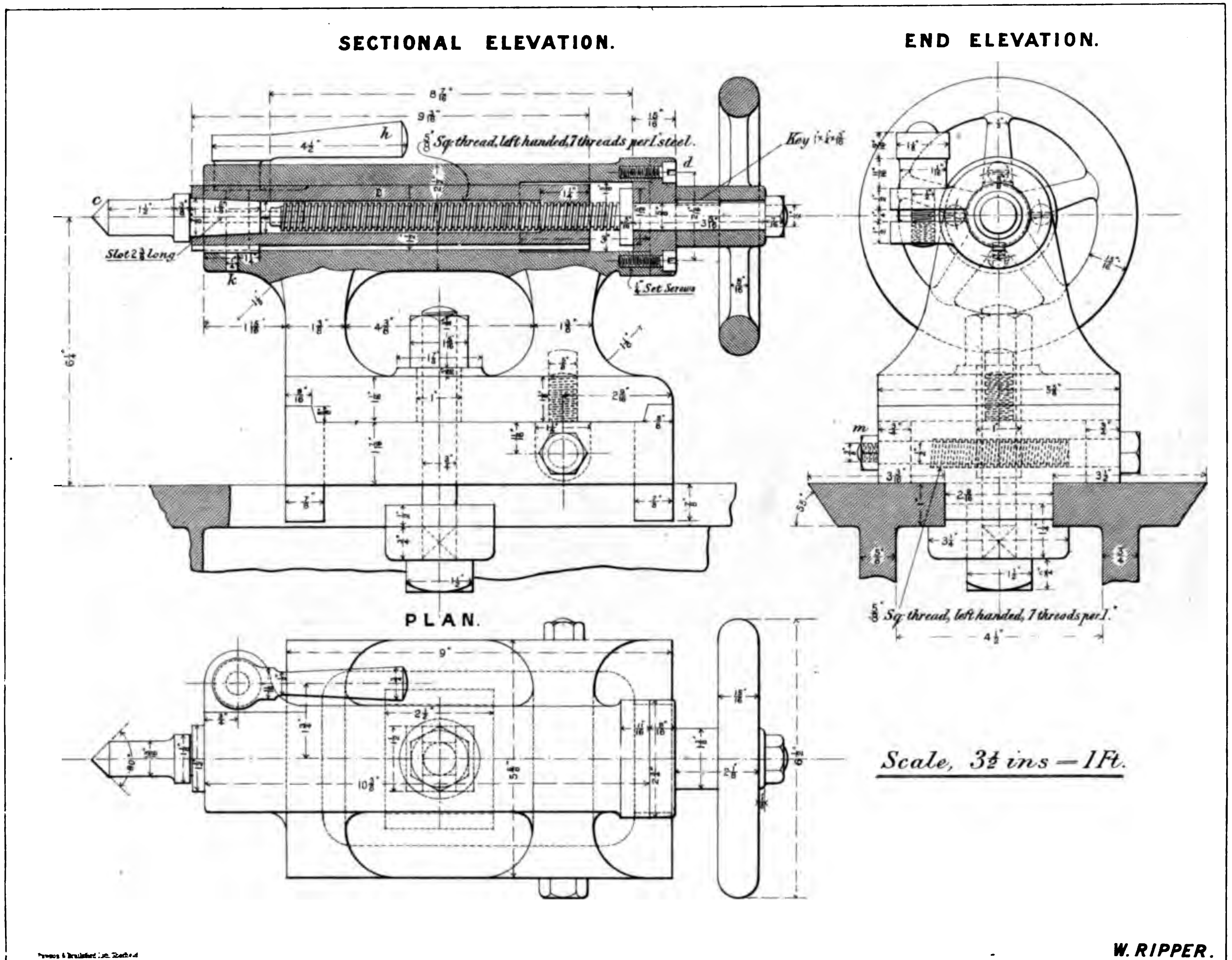
LS leading screw. BS back shaft. RA radial arm.

FAST HEADSTOCK.



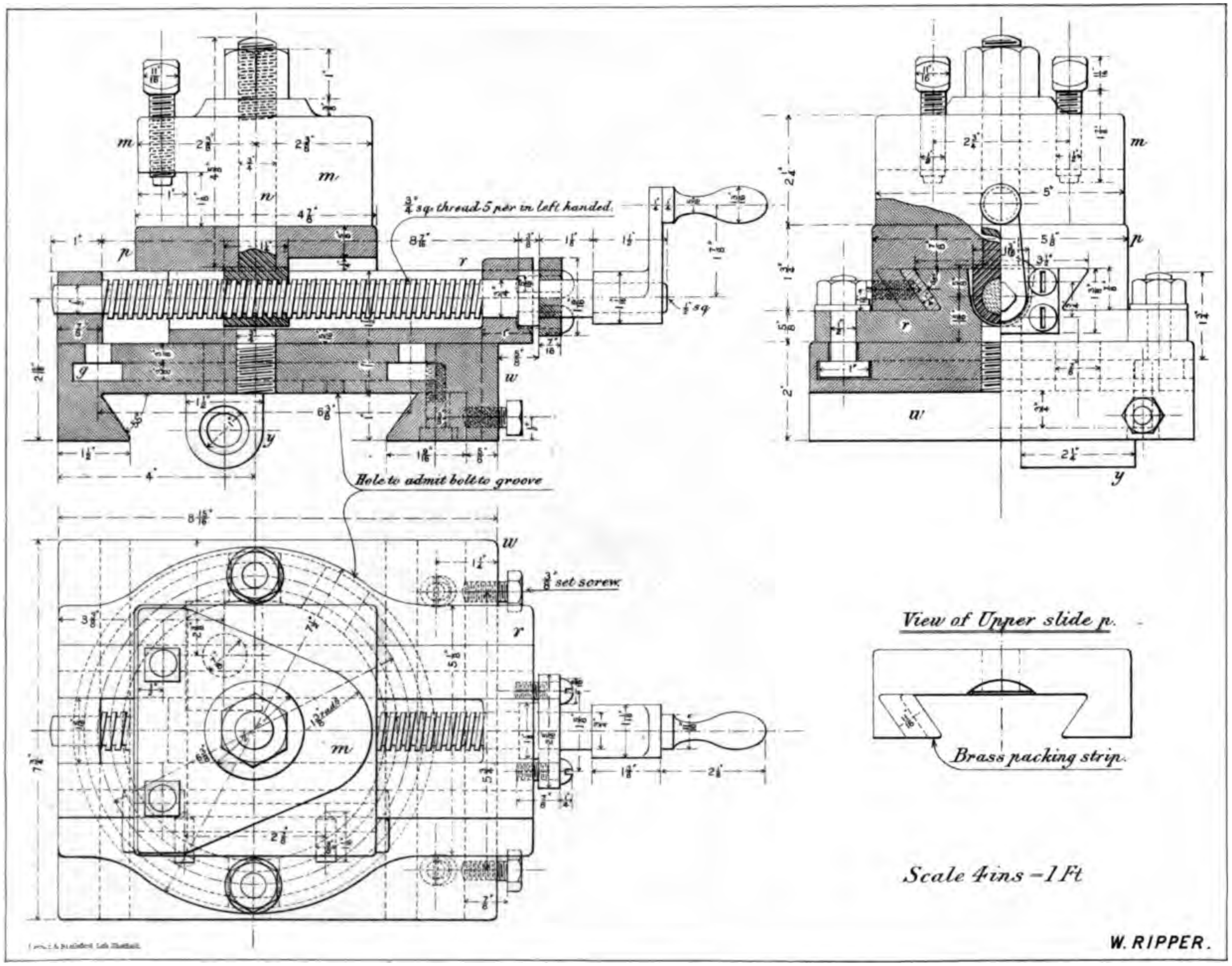
***M* lathe spindle or mandril. *FP* face plate. *C* centre. *SC* speed cone or stepped cone pulley. *H* back gear handle. *P* pin for securing back gear. *A* stud for securing swinging plate. *E F G K* wheels for reversing.**

LOOSE HEADSTOCK OR POPPET HEAD.



DETAILS OF COMPOUND SLIDE REST.

PLATE L



SLIDE REST DETAILS.